U. S. ARMY
TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

TREC TECHNICAL REPORT 61-38

CYCLOIDAL CAM TRANSMISSION

TASK 9R 38-01-020-04

Contract DA44-177-TC-651

January 1961

Revised June 1961

prepared by

WESTERN GEAR CORPORATION
SYSTEMS MANAGEMENT DIVISION
2600 EAST IMPER'AL HIGHWAY
LYNWOOD, CALIFORNIA

NOX

4424-61

Mr. W. A. Hudgins/pb/21528

HEADQUARTERS
U. S. ARMY TRANSPORTATION RESEARCH COMMAND
TRANSPORTATION CORPS
Fort Eustis, Virginia

TCREC-ADS 9R38-01-020-04

SUBJECT: Improved Mechanical Transmission System (Cycloidal Cam Transmission)

TO: See Distribution List

1. The subject report discusses the contractor's efforts in determining the feasibility of using the Cycloidal Cam Transmission for Army aircraft.

2. This Command concurs in the recommendations contained in paragraphs 1, 2, and 3, page 13, and the continuing program will be scheduled on this basis. Paragraphs 4, 5, and 6 recommend extension of studies; however, in the opinion of this Command, these additional studies will not profitably meet the objectives of this program.

3. This Command concurs in the conclusions made by the contractor.

FOR THE COMMANDER:

Robert B. Mercer
ROBERT B. MERCER
Captain, TC
Asst Adjutant

Approved by:
Wayne A. Hudgins
Wayne A. Hudgins
Project Engineer

Task 9R38-01-020-04
Contract DA44-177-TC-651
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CYCLOIDAL CAM TRANSMISSION

Trec Phase I Final Report

Prepared By
WESTERN GEAR CORPORATION
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LYNWOOD, CALIFORNIA

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FORT EUSTIS, VIRGINIA


IN PUT
CYCLOIDAL CAM REDUCER

FOREWORD

Phase I Final Report covers the feasibility study of Cycloidal Cam Transmissions for Army aircraft application. The contract for this study was designated as DA44-177-TC-651, and the commencing date was June 28, 1960. The principal engineers and scientists engaged in the study are T. Barrow, A.W. Brandstetter, J.S. Morgan and R.H. Tomren, from the Lynwood, California facility of Western Gear Corporation. The company contract administrator is Eric Marlor. The government representatives at U.S. Army Transportation Research Command Headquarters, Fort Eustis, Virginia, are Lt. Colonel A.M. Steinkrauss, Contract Administrator, and Mr. Wayne A. Hudgins, Project Engineer.

The contract is divided into three phases, namely Study, Design and Manufacture. The report covers the period June 23, 1960 through January 24, 1961, which is the study phase. This portion of the report is defined as an analysis of the capacity of the subject transmissions to provide Army aviation with a lightweight, high-reduction system design. The study, moreover, is to disclose the upper power transmission limitations of the concept relative to efficiency, fatigue life, reliability and other factors. It is limited to reduction ratio aspects, in the range from 18:1 through horsepower, and the speed range covers 4760 to 30,000 rpm. Special emphasis was given to the 280-2520 horsepower and 19:1-150:1 reduction ratio sections to study compatibility with present-day gas turbine engines.

During the work, the engineering team was aided by such government organizations as the U.S. Patent Office, the U.S. Government Research Reports and Armed Services Technical Information Agency (A.S.T.I.A.).

Grateful acknowledgement is extended to Professor Charles K. Wojcik of U.C.L.A. for his review of our kinematic studies; to Hyatt Bearings Division for their roller bearing material selections; to SKF Industries for their aid in assigning the distribution of the dynamic loads; to Remington Rand for their programming of cam geometry and to Douglas Aircraft Company for their aid in template grinding.

The advice and counsel of the following people at Western Gear Corporation was very beneficial: L. A. Acurso, E. T. Bergquist, M. E. Conlon, J. J. Davis, W. O. Engstrom, M. L. Headman, J. K. Morris, R. A. Muller, J. H. Nasmyth, L. F. Perron, F. Shortman, J. J. Webber. In addition, the following consultants aided the program: J. C. Hambric and L. D. Martin. There also remains the pleasant duty of thanking Hannchen Beard for the typing and proofreading of this report.

The conclusions herein have been arrived at through the theoretical analyses so far performed, but have not been test proven and should not be considered as final conclusions by the authors or Western Gear Corporation.

R. H. Tomren

January 1961
Revised Edition
June 1961
Lynwood, California

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LIST OF SYMBOLS

ϕ	=	Angular advance of the generating circle
a	=	Radius of epicycloid generating circle
R	=	Radius of base circle
b	=	Radius of epitrochoid generating circle
ρ	=	Radius of equidistant generating circle
n	=	$\frac{R}{a}$ = number of cam lobes and reduction
β	=	Angle subtended by the normal to the equidistant and a line through the center of the equidistant generating circle parallel to the line making the angle ϕ with the Y axis
γ	=	Vertical component of ρ
χ	=	Horizontal component of ρ
X	=	Horizontal offset of curve parametric in ϕ
Y	=	Vertical offset of curve parametric in ϕ
HP	=	Horsepower
T	=	Torque
s_c	=	Compressive stress
s_t	=	Tensile stress
s_s	=	Shear stress
s_{cs}	=	Combined stress
E	=	Young's Modulus
I	=	Moment of inertia

I_p	=	Polar moment of inertia
M	=	Bending moment
t	=	Section thickness
PSI	=	Pounds per square inch
H	=	Hertz stress
e	=	eccentricity
rpm	=	Revolutions per minute
L	=	Length
R, r	=	Radius of curvature (large, small)
P, F, y	=	Load or force
c	=	Extreme fibre distance
Z	=	Section modulus
x	=	Variable
v	=	Linear velocity
ω	=	Angular velocity
P	=	Dynamic load vector
P_n	=	Maximum dynamic load vector
α	=	Dynamic load vector angle
W	=	Weight
m	=	mass
g	=	Gravity acceleration
I. D.	=	Inside diameter

O. D. = Outside diameter

D = Bearing pitch diameter

C_F = Bearing friction coefficient

W_{BDC} = Full complement needle roller basic dynamic capacity

C = Bearing capacity at operating load and speed

S = Stroke

SUMMARY

During Phase I of the contract, weekly engineering meetings were held for the purpose of reviewing progress and to stimulate new ideas. Listed below are highlights of the meetings:

Meeting Number 1 July 5, 1960 Contract Briefing	A general review of the contract covering the three phases; Study, Design and Manufacture.
Meeting Number 2 July 12, 1960 Literature Search	Ground work for the method of tabulating literature was established. The method is based on TRECOM Circular 715-10. The range of transmission investigation was divided into three general areas: - 250 horsepower 30:1 to 150:1 ratios 800 horsepower 30:1 to 150:1 ratios 2500 horsepower 30:1 to 150:1 ratios
Meeting Number 3 July 18, 1960 Cycloid Cam Kinematics	The basic fundamentals of the cycloidal cam was reduced to a three-lobe cam. A paper and wooden model was started so that the action could be better understood. Planetary gearing, bearings, materials, stress, etc. were outlined for investigation in relation to the subject contract.

Meeting Number 4
July 26, 1960
Design Parameters with Model Studies

A review of a visit to TRECOM and the Patent Office delineated the paths for research and study in Phase I. Parameters of some of the current helicopter transmissions were reviewed for comparisons with cycloidal cam parameters. The method of gathering preliminary manufacturing information was started.

Meeting Number 5
August 2, 1960
Patent Evaluation and Design Sketches of Cam Action

Patent evaluation revealed that a combination of gearing and cycloidal cam action would be a good approach. The major portion of the literature search was completed. Cam action of the three-lobe study revealed the relationship of the eccentric movement relative to the fixed pins and reverter pins.

Meeting Number 6
August 9, 1960
Manufacturing Techniques for Minimal Costs

Grinding operations for the cam plates and other rolling surfaces were considered for surface hardness, tolerances, accuracies, etc.

Meeting Number 7
August 16, 1960
Weight and Loading Analysis vs Horsepower Spectrum

Four-bar-link vector analysis is one of the basic criteria for preliminary design layouts. The three cam plate configuration is preferred.

Meeting Number 8
August 23, 1960
Analytical Study for Ratios 28:1 through 160:1

Cam layouts for various ratios and sizes revealed the practical limits of width and depth of cam lobes. The information was reviewed with our production people for their comments and suggestions.

Meeting Number 9
August 30, 1960
Component Evaluation for Reliability

The multiplicity of bearings for the transmissions upsets the classical reliability theory. Modification of theory was reviewed in terms of fail-safe aircraft operations.

Meeting Number 10
September 6, 1960
Review of Material for Progress Report Number 2

The material included preliminary designs for 250 horsepower and 800 horsepower, as well as loading and design information.

Meeting Number 11
September 13, 1960
TRECOM visit

Wayne Hudgins of TRECOM visited Western Gear. The advantages of staging for ratios in excess of 30:1 were demonstrated by graphical layouts. UNIVAC 120 computer can be adapted for mathematical expressions. Radioisotope tracing could be a feasible method for studying wear of component parts.

Meeting Number 12
September 20, 1960
Open Discussion

TRECOM requested a Display Model be delivered to them by October 20th. Many ideas were offered and these were resolved into a working model (6144B27). Parametric data in the form of charts and graphs were completed and included in Progress Report Number 3.

Meeting Number 13
September 27, 1960
Review of Parametric Data

Programmed data for UNIVAC 120 has been resolved, and as an end product, a radius relationship was discovered for the cam lobes. This means that the cams can be manufactured and checked to a high degree of accuracy.

Meeting Number 14
October 4, 1960
Dry Weight Comparison of Gear and Cycloidal Cam Transmissions

Most of the meeting was devoted to the progress of the Display Model through its various manufacturing stages in the experimental department. A great deal of "model" interest has been generated. Dry weight values of .3 to .4 pounds per horsepower appear to be realistic.

Meeting Number 15
October 11, 1960
Final Design Review

Single staging, double staging, modularizing, differential, etc. were covered in the review.

Meeting Number 16
October 18, 1961
Final Report Review for Progress Report Number 3

Material will include model information as well as any recent new ideas.

Meeting Number 17
October 25, 1960
Work Assignments for Final Report

Work assignments were made for Phase I, Final Report.

Meeting Number 18
November 1, 1960
Final Meeting

The final report (rough draft) scheduled for delivery to TRECOM November 19th. More new ideas were crystallized for the report.

Listed below is a summation of information developed during Phase I:

1. The Cycloid Cam Transmission appears to be feasible for Army aircraft.

2. The combined dynamic and static load stresses are not considered excessive.

3. Use of exotic materials would enable the weight/horsepower ratio to be further depressed, if desired.

4. The weight/horsepower ratio for reductions in the 19:1 to 150:1 range for a spectrum spanning the interval from 280-2520 horsepower, inclusive, varies from .30 to .40.

5. The efficiencies are comparable with conventional gear systems in the low horsepower range, but better at the high end of the horsepower and reduction spectrums.

6. Reduction is a function of the ratios of the base circle radius to the generating circle radius or:

$$\frac{R}{r} = n$$

7. The crank eccentricity should not exceed 60-75 percent of the generating circle radius for workable profiles.

8. Maximum crank eccentricity should be limited to hold crank bearing loads to nominal values.

9. Reaction rollers rotate with a non-uniform rpm and in the same direction as the input shaft.

10. The outer races of the reverter bores, which are integral with the cam plate, rotate about the reverter pin with the same frequency and in the same sense as the input shaft.

11. The lines of action of the roller loads all converge at the instantaneous center of the system. The instantaneous center lies along the line made by the crank center and the axial center of the input at a distance ℓ from the latter, where:

$$\ell = e' + \bar{e}_x \frac{\omega_1}{\omega_2}$$

e = eccentricity

ω_1 = input rotary speed

ω_2 = output rotary speed

12. The dynamic load acts along the line of eccentricity and through the point of confluence of the reaction roller loads. The magnitude of the dynamic load per cam plate is:

$$F = me\omega_i^2$$

Where:

m = mass of the gyrating cam plate assembly

c = crank eccentricity

ω_i = input rotary speed

13. Many practical cam lobe layouts disclose the profiles to be circular arcs.

14. The significance of item 13 is that manufacture and inspection of the cam lobes is simplified.

15. The general expressions for the cam profile, in parametric form, have been programmed for Western Gear's UNIVAC 120. The expressions are:

$$X_c = (R+a)\mathrm{Sin}\,\emptyset + b\,\mathrm{Sin}\left[\frac{(R+a)\emptyset}{a}\right] - \rho\,\mathrm{Sin}\left[\mathrm{Tan}^{-1}\left(\frac{b\,\mathrm{Sin}\left\{\frac{Rx\emptyset}{a}\right\}}{a+b\,\mathrm{Cos}\left\{\frac{Rx\emptyset}{a}\right\}}\right) + \emptyset\right]$$

$$y_c = (R+a)\mathrm{Cos}\,\emptyset x b\,\mathrm{Cos}\left[\frac{(R+a)\emptyset}{a}\right] - \rho\,\mathrm{Cos}\left[\mathrm{Tan}^{-1}\left(\frac{b\,\mathrm{Sin}\left\{\frac{Rx\emptyset}{a}\right\}}{a+b\,\mathrm{Cos}\left\{\frac{Rx\emptyset}{a}\right\}}\right) + \emptyset\right]$$

16. The load studies were based on the 4-bar linkage principle.

17. The bearing industries practices were reviewed for allowable Hertz stresses.

18. The Hertzian S-N curves were charted and used for determinations of cyclic life for bearing surfaces.

19. A display model with an 18:1 reduction was constructed.

20. The display model verified the system kinematics.

21. Inversion of the cycloid was practiced with the model. The tabulated results follow:

Type	Drive Member	Stationary Member	Driven Members	Input/Output Ratio	Output Rotation Referred to Input
A	eccentric	reaction cage	reverter cage	N	Opposite
B	eccentric	reverter cage	reaction cage	N+1	Same
C	reverter cage	eccentric	reaction cage	$\frac{N}{N+1}$	Same
D	reaction cage	eccentric	reverter cage	$\frac{N+1}{N}$	Same

22. A 280 horsepower module has been presented in two preliminary designs. One is a 19:1 and the other a 29:1 reducer.

23. These units serve as the building block modules for greater capacity units, covering the power spectrum from 280-2520 horsepower and reduction ratios from 19:1 to 150:1. Illustrations of the nested assemblies covering the horsepower spectrum have been made.

24. The preliminary designs have been developed to assure the frontal area diameters fall within those of the jet engines they are to complement.

25. Bearings were selected for a design B-10 life of 1000 hours.

26. The two main structural materials recommended are steel and aluminum. The steels are AISI No. 4340, 4620, 9310 and 52100. The aluminums are 7075T6 and 356T6.

27. Exotic materials have been selected as alternates for those already listed. Their use would increase prototype costs. Secondary consideration is offered them due to their lower specific gravities. The materials are Titanium, to replace steel and magnesium as a substitute for aluminum. The

Titanium alloys are: Ti-5A1-2.5 Sn, Ti-8A1-1 MO-1V and Ti-6A1-4 Zr-1V. The magnesium alloys are: A-Z63 and A-Z92. Beryllium alloys, judiciously chosen, are also applicable.

28. The inversion characteristic of the cycloid makes for flexibility and greater reductions when staged.

TYPE		E	F	G	H
MEMBER					
1st Stage					
	DRIVE	Eccentric	Eccentric	Eccentric	Eccentric
	STATIONARY	Reaction Cage	Reaction Cage	Reaction Cage	Reaction Cage
	DRIVEN	Reverter Cage	Reverter Cage	Reverter Cage	Reverter Cage
2nd Stage					
	DRIVE	Eccentric	Eccentric	Reverter Cage	Reaction Cage
	STATIONARY	Reaction Cage	Reverter Cage	Eccentric	Eccentric
	DRIVEN	Reverter Cage	Reaction Cage	Reaction Cage	Reverter Cage
REDUCTION RATIO		N_1N_2	$N_1(N_2+1)$	$\frac{N_1(N_2+1)}{N_2}$	$\frac{N_1N_2}{N_2+1}$
OUTPUT ROTATION REFERRED TO INPUT		Same	Opposite	Opposite	Opposite

29. The modularized design proposed by the study is compatible with parts standardization.

30. Maintenance benefits from modularization permits sub-assembly removals and replacements.

31. Studies have been made for accessibility to the high and low speed output shafts of jet engines.

32. Parametric charts and graphs have been drawn for the modular cycloid reducers to emphasize their attractive features and/or to compare them with contemporary designs. They are:

    Frontal Area vs Horsepower
    Volume vs Horsepower
    Weight vs Horsepower
    Weight/Horsepower vs Input Speed
    Weight vs Input Torque.

33. The cost estimates for the experimental units are based on preliminary designs. The cost estimates are presented in different breakdowns as follows:

    Cost/Pounds vs Weight
    Cost/Pounds vs Horsepower
    Tooling and Test Costs vs Horsepower
    Overall Cost vs Horsepower.

34. The Cycloid Cam Transmission studies have been extended beyond the modular and the single stage concepts and embrace a Roller Cam and a Differential Cam configuration.

35. Both the latter investigations involve transmission capabilities of 280, 840, 1680 and 2520 horsepower and in each horsepower level studies of reduction ratios of 29:1, 59:1, 89:1 and 149:1.

36. The Single Stage Roller Cam study results were tabulated and graphed. the graphs reflect:

    Weight vs Horsepower
    Housing Diameter vs Horsepower
    Weight vs Output Torque
    Drive-Dynamic Loads vs Ratio (horsepower parameter)
    Reduction
    Ratio
    Parameters.

37. The Differential Cam Transmission characteristics were reduced to tabular form and incorporated in graphs representing:

Weight vs Ratio
Housing Diameter vs Ratio
Horsepower Parameters.

38. A kinematic study of the differential cam action was made and illustrated.

39. Single stage cycloid cam transmissions of reduction ratios 29:1, 59:1, 89:1 and 149:1 for 280, 840, 1680 and 2520 horsepower units were analyzed. The results were tabulated and graphed.

Cam Plate Diameter vs Horsepower
Weight vs Output Torque
Weight vs Horsepower
with reduction ratio parameters.

40. A preliminary Cycloid Roller Cam layout for an 840 horsepower, 29:1, single stage reduction unit was made.

41. Some of Western Gears' participation and experience in military aircraft transmission programs has been compiled and tabulated.

42. A patent search was instituted, and the material reviewed was qualitatively evaluated. More than thirty patents were carefully scrutinized. All were primarily concerned with high-ratio, single stage reducers. The tabulated results are included.

43. An extensive bibliography has been compiled and included herein.

CONCLUSIONS

The Cycloidal Cam Transmission appears feasible for Army aircraft. The attributes that assure feasibility are:

1. The weight/horsepower ratio appears to be lower than for contemporary transmissions of comparable reduction.

2. Uninterrupted engagement of the working surfaces contributes to smooth, quiet operation and minimizes dynamic shock loading. The reduction system backlash requirements are low as compared to those required for conventional gearing.

3. Efficiencies are comparable with low ratio spur gear systems. The high ratio planetary gear systems (usually employed in contemporary transmissions of comparable reduction) appear to be less efficient than the Cycloidal Cam Transmission.

4. Dynamic balance of the cam system by the utilization of the triple cam plate design should permit high input speeds of modern prime movers. Such speeds are not practical for transmissions using a single or double plate cam system.

5. The crank eccentricity must be maximized to restrain crank bearing loads to practical values. A safe maximum is 60-75 percent of the cam generating circle radius.

6. The cam lobe profiles are arcs of circles. This permits accurate generation of profiles.

7. Cam-cutter diameter should be the same as the diameter of the reaction rollers.

8. Frontal-area diameters of the modular configuration are compatible with existing turbine engines of comparable horsepower.

9. The modular configuration has an important advantage. Where the use of two or more similar modules are used to handle a range of higher horsepower capacities, the logistic problems and costs are reduced. Where servicing is required, a serviced and tested module could be substituted for a defective unit. This should be of particular importance in advanced base operations.

10. The Differential Cam Transmission displays a more favorable frontal-area diameter with varying horsepower and reduction ratios than either of the single-stage cycloids or the roller-cam configurations.

11. The differential cam drive weight curves show a pronounced upward trend with decreasing reduction ratios.

12. The compatibility of the differential cam concept is more suitably disposed toward the high, rather than the low, reduction ratios.

13. The reduction ratio for the single-stage, differential cam drive, with eccentric shaft input and reaction roller cage output is:

$$R = \frac{N_1(N_2+1)}{N_1-N_2}$$

Where:

N_1 = number of cam lobes on 1st-phase cam

N_2 = number of cam lobes on 2nd-phase cam

a. When $N_2 < N_1$, input and output rotate in the same direction.

b. When $N_2 > N_1$, input and output rotate in opposite directions.

RECOMMENDATIONS

1. In the interest of simplicity, and to reduce manufacturing time and costs, it is recommended that a finalized design of the basic 280-horsepower module be processed. The ratio could be either 19:1 or 29:1, but preferably the former, since this most closely approximates present transmissions used to drive rotating-wing vehicles.

2. Work on the module design of the 840-horsepower transmission (6144D51) should be deferred until the refined 280-horsepower module design has been completed and analyzed. The findings should result in more compact and less costly future modules, and at the same time establish parameters for the design of modules of greater horsepower.

3. It is recommended that a 50-hour torque load test be conducted on the 280 horsepower module during Phase III of the contract. For comparison with the noise levels of present geared transmission, sound level studies should be made during the test program.

4. The experience, gained in manufacturing the display model, led to the conclusion that it is possible to relax some of the manufacturing tolerances applied to the cam lobes. Furthermore, it may not be mandatory to grind the cam lobe contacting surfaces. The physical dimensional studies should be made of these areas to determine the effect of increased cam lobe to fixed roller clearances and to establish these limits. This may disclose that un-ground cam profiles will be satisfactory.

5. We recommend that a model be made to study the kinematics of the right-angle cam transmission.

6. Further experimentation with the present Display Model

is recommended. Below are a few suggestions:

a. Replace the lucite cam plates with metal cam plates and conduct simple loading tests.

b. Attach "strainline photoelastic" gages to the cam plates in order to observe the loading kinematics during testing.

c. Installation of anti-friction bearings and/or sleeve bearings so that a momentary input speed of 6000 RPM may be observed.

d. Take slow motion movies of the model during the experimental testing.

STATEMENT OF THE PROBLEM

Problem Areas

The problems involved in high ratio reduction units resolve into two basic groups:

(1) static loading characteristics,
(2) dynamic performance.

In static loading, stress, deflection, direction of forces and reactions, interaction of moving members with respect to each other and the housing must all be evaluated, if a minimum weight unit with maximum efficiency and capacity is to be achieved. Bearing loads, compression loads at reaction points, shearing and bending loads on reaction members, shafts and housings must all be considered.

Consideration of dynamic performance raises the perennial problem of combining high speeds at the input with high loads in the output for a single stage. One approach to relieve this situation is the use of anti-friction bearings. Load concentration and space required for anti-friction bearings in the quantities necessary, in some types of reducers, severely limit the transmission arrangement. Another solution is to design for low relative movement per revolution of the input at the reaction points so that the total heating is low.

In some designs of high ratio reducers, wear and consequent enlargement of clearances increase the pressure angles, thus aggravating the loading. An error in spacing, or radial distance of reaction members, or any combination thereof, result in concentrating load on one or more points, and under high input speeds results in localized shock loading. The condition is aggravated by a high ratio requiring a large number of individually located pins, holes, slats, plungers, etc. The precise location of parts increases fabrication and inspection costs proportionately.

Unbalance and vibration are serious problems with high speed inputs, and since most high ratio, single stage reductions operate with eccentric cam inputs and with differential output, the vibration magnitude increases with speed and cam offset. Errors in progressive action create vibration of a lower amplitude and higher frequency which are both noisy and destructive.

The ideal, toward which all high-ratio, single stage reductions strive, is for high loads at the point of no-motion, and high speed at the point of no-load. Unfortunately, successive wedging or lever action which creates the force to move the output member must engage all the teeth, lobes, pins, plungers, etc., of whatever system is in use, at a rate corresponding to the input speed rather than the output speed, and while the relative sliding velocity at each engagement can be made quite small, the total sliding distance under output reaction loads represents a considerable power loss. The possibilities for the greatest gain in a high ratio reduction lie in determination or discovery of methods to reduce this inherent loss in differential action.

DISCUSSION

INTRODUCTION

The Cycloid Cam Transmission has attributes which are different than those found in conventional gear systems. It incorporates an eccentric input; a plate whose peripheral surface is continuously profiled with a sequential series of identical cam lobes, reaction pins and reverter pins. The aforementioned drive elements remain in constant engagement, whereas with gears the load is carried by a relatively few number of teeth in mesh at any given instant. The absence of mesh interrputions as occasioned in gear transmissions, eliminates noise, heating, backlash and roughness and inhibits the generation of destructive dynamic loads. The energy conserved is reflected in the greater efficiencies made possible with Cycloid Cam Transmissions.

There is another more fundamental reason for the basic superiority of the cycloid cam principle. The cycloid permits only pure rolling action because it is described by a point in the plane of one circle as it rolls on another.

The study effort is designed to demonstrate the preferable characteristics of the cycloid cam drive. The study begins with a review of roulettes, of which generic family the epitrochoid is a member, and from which the equidistant curve is generated. The equidistant curve is the actual profile of the cam lobe surfaces of the cycloid cam drives.

Kinematics of Trochoids and Roulettes

A roulette is a path made by a point located in the plane of one curve as that curve rolls on another. The number of possible roulettes is infinite. Those mostly used in engineering practice are the trochoid and the involute (of which the cycloid and cardiod are special cases).

All roulettes made by a point in the plane of one circle as it rolls on another circle are termed trochoids. When the tracing point is in the circumference of the rolling circle, the trochoids, because of their general usefulness are specifically known as:

Epicycloid, when the generating circle rolls on the outside of the fixed circle.

Cycloids, when rolling on a straight line.

Hypocycloids, when rolling on the inside of a fixed circle.

Pericycloids, when the rolling circle is larger than the fixed circle and the inside of the rolling circle rolls on the outside of the fixed circle.

The Epitrochoid is of particular interest in this study. The Prolate Epitrochoid, a special case of the Epitrochoid, in which the tracing point lies within the circumference of the generating circle, is adaptable to the design of multilobed cycloidal cams. Its adaptability is attributable to its lack of cuspiness, inherent in the epicycloid, and its freedom from reversion, characteristic of the Curtate Epitrochoid.

Drawing 6144D46 graphically portrays the generation of the epicycloid. The layout displays a six-lobed cam on which two lobes are fully developed. The relationship between base circle, generating circle and number of lobes is:

$$\frac{R}{a} = n$$

Where:

R = Base circle radius

a = Generating circle radius

n = number of cam lobes and reduction

Generation of the Equidistant Curve

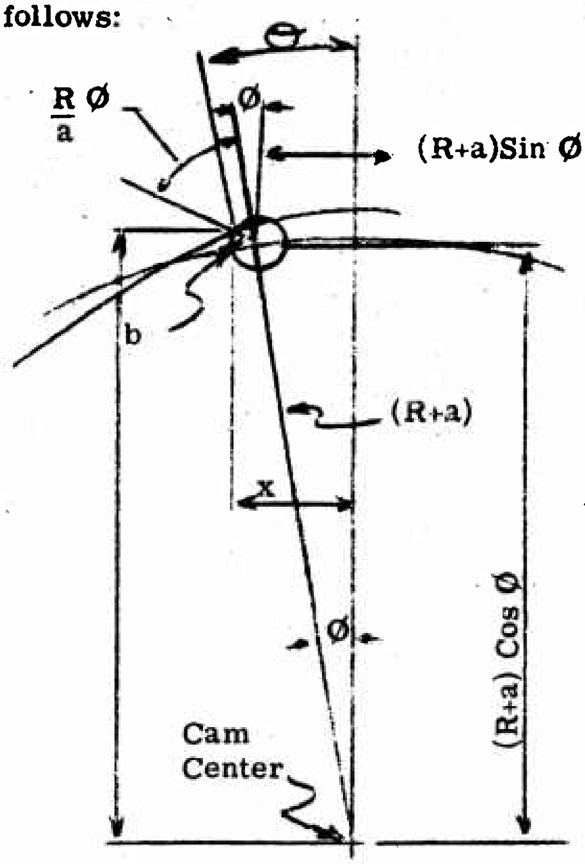
The epicycloid begins and ends abruptly at the base circle in a discontinuous cusp. A roller of any diameter other than a point could not traverse this path. The epitrochoid is more adaptable in this sense as it can accommodate a limited finite pin size. When the design requires a heavier pin than the compliance of the epitrochoid can effect, resort must be made to the equidistant curve. The equidistant is generated by making the epitrochoid the centrode for a system of circles equal in diameter to the required pin size. Drawing 6144D46 illustrates the relationship between these curves and the number in which they are generated.

The equation of the Prolate Epitrochoid in parametric form is:

$$x_p = (R+a)\ \text{Sin}\ \emptyset + b\ \text{Sin}\ \left(\frac{R+a}{a}\right) \emptyset$$

$$y_p = (R+a)\ \text{Cos}\ \emptyset + b\ \text{Cos}\ \left(\frac{R+a}{a}\right) \emptyset$$

The derivation follows:



$$x = (R+a)\ \text{Sin}\ \emptyset + b\ \text{Sin}\ \left(\frac{R}{a}+1\right) \emptyset$$

$$y = (R+a)\ \text{Cos}\ \emptyset + b\ \text{Cos}\ \left(\frac{R}{a}+1\right) \emptyset$$

Where:

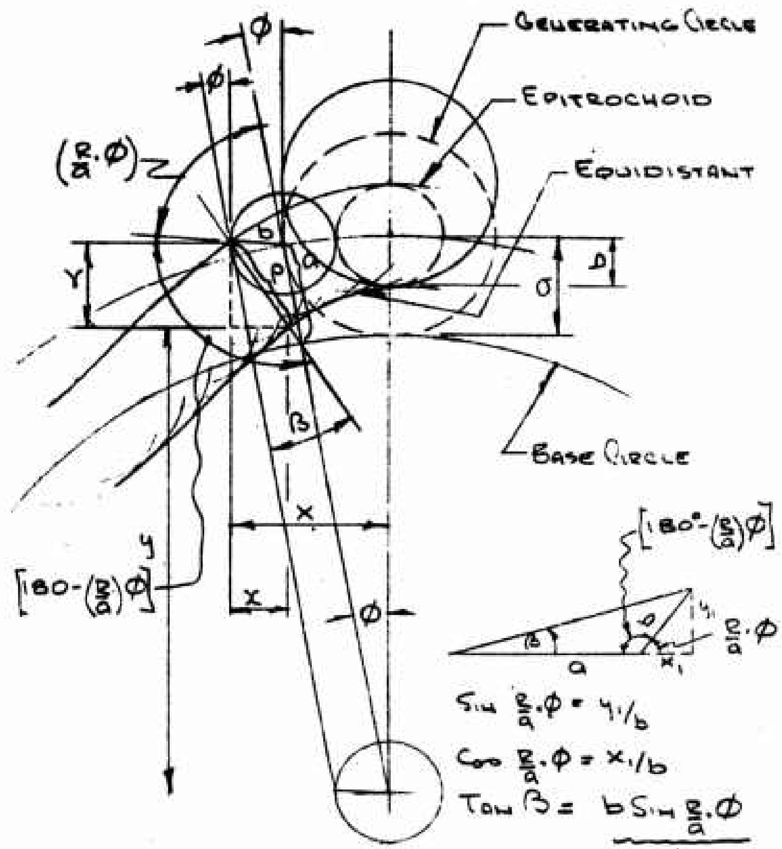
R = Base Circle Radius
a = Generating Circle Radius
b = Tracing Circle Radius
n = R/a = Number of lobes

The equation of the equidistant curve in parametric form is:

$$x_e = (R+a)\sin\phi + b\sin\left[\left(\frac{R+a}{a}\right)\phi\right] - \rho\sin\left[\tan^{-1}\left(\frac{b\sin\left\langle \frac{R}{a}\cdot\phi\right\rangle}{a+b\cos\left\langle \frac{R}{a}\cdot\phi\right\rangle}\right) + \phi\right]$$

$$y_e = (R+a)\cos\phi + b\cos\left[\left(\frac{R+a}{a}\right)\phi\right] - \rho\cos\left[\tan^{-1}\left(\frac{b\sin\left\langle \frac{R}{a}\cdot\phi\right\rangle}{a+b\cos\left\langle \frac{R}{a}\cdot\phi\right\rangle}\right) + \phi\right]$$

The derivation follows:



$$\sin\frac{R}{a}\cdot\phi = y_1/b$$

$$\cos\frac{R}{a}\cdot\phi = x_1/b$$

$$\tan\beta = \frac{b\sin\frac{R}{a}\cdot\phi}{a+b\cos\frac{R}{a}\cdot\phi}$$

$$\beta = \tan^{-1}\left[\frac{b\sin\frac{R}{a}\cdot\phi}{a+b\cos\frac{R}{a}\cdot\phi}\right]$$

$$\gamma = \cos(\beta+\emptyset)$$

$$\chi = \sin(\beta+\emptyset)$$

$$X_e = X-\chi = (R+a)\sin\emptyset+b\sin\left[\frac{(R+a)\emptyset}{a}\right]-\rho\sin(\beta+\emptyset)$$

$$Y_e = Y-\gamma = (R+a)\cos\emptyset+b\cos\left[\frac{(R+a)\emptyset}{a}\right]-\rho\cos(\beta+\emptyset)$$

Then

$$X_e = (R+a)\sin\emptyset+b\sin\left[\frac{(R+a)\emptyset}{a}\right]-\rho\sin\left[\tan^{-1}\frac{\left(b\sin\left\langle\frac{R\emptyset}{a}\right\rangle\right)}{a+b\cos\left\langle\frac{R\emptyset}{a}\right\rangle}+\emptyset\right]$$

$$Y_e = (R+a)\cos\emptyset+b\cos\left[\frac{(R+a)\emptyset}{a}\right]-\rho\cos\left[\tan^{-1}\frac{\left(b\sin\left\langle\frac{R\emptyset}{a}\right\rangle\right)}{a+b\cos\left\langle\frac{R\emptyset}{a}\right\rangle}+\emptyset\right]$$

Kinematics of The Reaction and Reverter Rollers

Drawing 6144D46 develops the centrode of the reverter bores and the locus of the contact between reverter pin and bore. The lines joining the corresponding reverter bore centers and contact points as the cam advances demonstrate the rotation of the reverter bore about the reverter pin center. The reverter bore rotates about the pin center in the same sense and in synchronism with the input.

In like manner the reaction rollers rotate at a non-uniform angular velocity in a clockwise rotation or more generally in the same sense as the input. Thus as the cam migrates from one pin to another the reaction rollers and reverter bores each complete a full revolution with respect to ground but in a contra direction to the cam.

DRIVE LOAD ANALYSIS

As the input is a uniform rotary motion so is the contra rotation of the cam plates. Every full rotation of the eccentric imparts to the cam plate a primary motion in one direction about a circle of radius e (eccentricity). This is accompanied by a secondary motion of counter-rotation or retardation through an angle equal to:

$$\emptyset = \frac{(n_p - n_L)}{(n_L)} \times 2\pi$$

Where:

n_p = number of pins

n_L = number of cam lobes

$\emptyset$ = angle of contra rotation

The meshing of the cam lobes and reaction pins ensures a hypotrochoidal action of the cam plate reverter bores, to provide an exact ratio of reduction between the angular rotation of the drive shaft, and the secondary rotation of the cam plates. The total motion of the cam plate is a combination of the primary and secondary motions, and these are transformed into the uniform rotary motion of the output by the reverter pins. The study is simplified by employing a system of four-bar kinematic linkages operating together or in parallel as illustrated in Figures 1 and 2 on Drawing 6144D47.

Figure 1 depicts the system for the interaction of one pin with instantaneous center at c. The Force F at radius e provides the turning moment to be resisted by the forces F_1, F_2, F_3, etc. with moment arms CR_1, CR_2, CR_3, etc. The magnitudes of these forces are inversely proportional to their moment arms. By the resolution of the forces around the pins, all the loads normal to the reverter pin arms are found to be of equal magnitude. This is a necessary and sufficient condition to insure a uniform rotary output under load.

The Cycloid Cam theoretically contacts all the reaction rollers. Consequently, there is a system of four-bar kinematic chains operating simultaneously and in parallel with the one of Figure 1. For each system the force contribution at the pins P is equal, but not necessarily the same as that for any other chain. Therefore the sum of all the forces at all the pins must be equal. The resulting torque at the pins must then equal the eccentric input torque.

To expedite the analysis more directly, it is well to consider Figure 2 of Drawing 6144D47, wherein the systems of four-bar linkages are being operated in parallel. Each reverter pin center shares the load in equal proportion, as demonstrated earlier. This approach again permits pin load analysis, and, in addition, a resolution of the bearing loads at C_0, C_1, C_2, etc.

Due to the difference between the number of pins and the number of cam lobes the load distribution is a function of the geometry. Consider pin P_6' in Figure 2 of Drawing 6144D47.

$$F_6 = \frac{T}{6} \times \frac{1}{OP'_6}$$

$$T = F \times e$$

The force along $P_6'R_6'$ is readily determined. In like manner the force in member $R_6'C_6$ may be evaluated. Then, at C_6 the bearing load can be resolved, knowing the magnitude and direction of one, and the directions, or lines of action, of the other two.

The bearing reaction at C_3 and C_4 are a little less obvious. In this case, the resisting moments $F_3 \times R_{(3,4)}$ and $F_4 \times R_{(3,4)}$ are first equated to T/6:

$$\frac{T}{6} = (F_3 + F_4)\ R_{(3,4)}$$

Where:

$$F_3 = F_4$$

$$F_3 + F = \frac{T}{6} \times \frac{1}{R_{(3,4)}}$$

Establishing $(F_3 + F_4)$ permits the further resolution of the forces along $C_3R_{(3,4)}$ and $C_4R_{(3,4)}$. The loads at the bearing points C_3 and $C_{(3A)}$ will now submit to analysis as again at each point one load is known in size and direction and the action lines of the other two are fixed.

Dynamic Load Analysis

The dynamic loading is occasioned by the flywheel effect of the rotary and translating masses. The total energy of the system at any instant is the sum of the energies of the rotating and translating parts. The various energies embodied in the system are:

$$E_1 = 1/2mv_1^2 \rightarrow [\text{Cam plates, eccentrics(main and reverter)}]$$

$$E_2 = 1/2I_1\omega_1^2 \rightarrow [\text{Cam plates, eccentrics(main and reverter)}]$$

$$E_3 = 1/2I_2\omega_2^2 \rightarrow [\text{Cam plates}]$$

$$E_4 = 1/2I_3\omega_3^2 \rightarrow [\text{Reaction Rollers}]$$

Total Energy

$$K_1E_1 = 1/2mv_1^2 + 1/2I_1\omega_1^2 + 1/2I_2\omega_2^2 + 1/2I_3\omega_1^2 = \underset{T/e}{F} \times \underset{2\pi e(\text{rpm})}{L}$$

Differentiating

$$\frac{d}{dt}\left[1/2mv_1^2 + 1/2I_1\omega_1^2 + 1/2I_2\omega_2^2 + 1/2I_3\,\omega_1^2\right] = \frac{d}{dt}(F \times L)$$

$$mv_1\frac{dv_1}{dt} + I_1 \quad {}_1\frac{d\omega_1}{dt} + I_2\omega_2\frac{d\omega_2}{dt} + I_3\omega_1\frac{d\omega_1}{dt} = F\frac{dL}{dt}$$

Since ω_1 and ω_2 are constant

$$\frac{d\omega_1}{dt} = \frac{d\omega_2}{dt} = 0$$

and

$$mv_1\frac{dv_1}{dt} = Fv_1$$

Whereby:

$$F = m\frac{dv_1}{dt}$$

Or

$$F = \frac{mv_1^2}{\bar{e}}$$

Thus the dynamic load is directed along the line of eccentricity and is equal to the total mass affected with the acceleration. (v_1) is the instantaneous linear velocity of the input crank eccentric.

Figure 3, Drawing 6144D47, illustrates the graphical solution for the determination of the instantaneous center of the system as a function of its inherent inertia when operative. A line joining the input and output velocity vectors a and b respectively, intersects the line of action of the dynamic load at the instantaneous center. Through this point pass all the action lines of the cam and roller bearing reactions.

Analytically, the same result is established quite simply by the relation.

$$e\,\omega_1 = R_0\,\omega_0$$

Where:

e = eccentricity

ω_1 = input angular velocity

ω_0 = output angular velocity

R_0 = $\overline{OP}_3 - e$

The distribution of the dynamic load over the supporting reaction rollers is: -

$$\Sigma\, P_{\gamma i} = P_n \Sigma \cos^{3/2} \gamma_i = F \text{ (known)}$$

$$P_n = F / \Sigma \cos^{3/2} \gamma_i$$

$$P_{\gamma i} = \frac{F}{\Sigma \cos^{3/2} \gamma_i} \cos^{3/2} \gamma_i$$

Consequently, the total load on the respective reaction rollers is the sum of the drive load and the dynamic load. This total load, converted into Hertz stress, determines the adequacy of the load contact area.

The load distribution at the bearing points, being a function of the geometry, has a fluctuating characteristic around the cam periphery as shown in Drawing 6144R17 for the 30:1 reduction. Simpson's Paraboloidal Rule applied to the load spectrum provides a weighted average equivalent load for bearing selection or design, compatible with the life and speed requirements. Shown on the same drawings are the reverter pin bearing loads, which display a similar fluctuating characteristic over the full complement employed. Bearing recommendations on these weighted averages, evaluated by Simpson's approximation serve also to establish these required capacities.

Simpson's Paraboloidal Approximation has the form:

$$\bar{Y} = \frac{x}{3n} \; y_0+4y_1+2y_2+4y_3+ \text{---} y_k$$

Where:

$x = 1$

y_0, y_1, y_2, etc. = Load ordinates at $k = 0, 1, 2$, etc.

$n = k_n$ where n is even

DESIGN CRITERIA

The design criteria governing the kinetics and kinematics of cycloids, are not as well defined analytically, or empirically, as more conventional systems. The resort to graphic analyses in the earlier part of the context, and its substantiation by line studies, validated the results. It was necessary to reduce the cam to a system of four-bar kinematic chains in order to resolve both the torque and dynamic loads. Drawing 6144D47 illustrates the procedure.(Figures 1, 2 and 3).

The reduction is established from the relation

$$\frac{R}{a} = n$$

Where:

R = base circle radius
a = generating circle radius
n = number of cam lobes and reduction.

The eccentricity e provides good equidistant cam surfaces when:

$$e = (60\% - 75\%)a$$

e determines the crank bearing load.

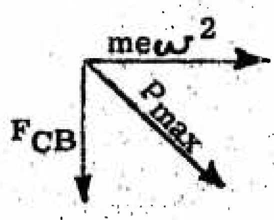
$$F_{CB} = \frac{HP \times 63000}{rpm \times e}$$

Since the center cam has twice the section of the end plates, the eccentric bearing load per plate is defined as:

$$\frac{F_{CB}}{4} = \frac{HP \times 63000}{4 \times rpm \times e}$$

For the units under consideration, this value should run to approximately 3000 pounds per single cam plate.

The crank bearings also support the dynamic load. Hence the combined load is:



Where:

e = eccentricity

ω = input ($2\pi \times rpm$)

$m = \frac{W}{g}$

The drive loads on the Reaction Rollers are established by the four-bar kinematic chain, as per Figure 1 of Drawing 6144D47. The input dynamic loads on the Reaction Rollers are proportioned in conformance with Figure 3, on the same drawing.

The adequacy of the cam and bearing surfaces must be established by an investigation of the Hertz stress, where:

$$Hertz_{max} = \left[.175 \times E \frac{\times P_{max} (1/R + 1/r)}{L} \right]^{1/2}$$

For the cam surface, P_{max} is equal to the highest load encountered

Where:

R = Radius of curvature at point of contact
r = Radius of reaction roller
E = Young's Modulus
L = Roller length

For the Reaction Roller needles and inner race

$$P_{max} = \frac{5 \times P}{n}$$

and where:

R = Inner race radius
r = Needle roller radius
L = Roller length
P = Total load supplied by roller
n = Number of rollers

In both cases, the maximum Hertz Stress should not exceed 324,000 psi.

The reaction and reverter needle roller bearing capacities must be predicated on a weighted radial load. The load spectrum for the reverters and reaction rollers provide weighted average values when Simpson's Paraboloidal Approximation is applied.

Superimposed on the loading of the reverter eccentric bearings, in addition to the transferred input drive, is the dynamic effect of the unbalanced eccentric reverter masses rotating at input speed. This load is represented by:

$$F_{ED} = me\,\omega^2$$

Where:

$$m = \frac{\omega}{g}$$

e = eccentricity

$\omega = 2\pi$ x rpm

where:

the rpm is the input speed.

The analytical section of this discussion describes the rotation of the reaction rollers, and the reverter eccentrics, as non-uniform and uniform respectively, although each makes one revolution for each revolution of the input. Despite the resolution of the required bearing capacities, on a weighted average load basis, it is necessary to re-examine the bearing design for maximum Hertz stress. The figure of 324,000 psi must not be exceeded. If design compromises cannot be effected to comply with this criterion, recourse must be made to a material such as Consutrode 8620, which invests bearing design with 336% of the capacity of conventional materials, authority of the Hyatt Roller Bearing Company.

The foregoing material establishes all the design parameters unique to the cycloid drive. The adequacy of all supporting componentry is determined by the usual stress analysis pattern.

In review, the design criteria might be outlined as follows:

Crank bearing load

$$F_{CB} = \frac{HP \times 63000}{rpm \times e}$$

Reduction, number of cam lobes and determination of physical size of base circle.

$$\frac{R}{a} = n$$

Where:

$$a = \frac{e}{(60\%-75\%)}$$

and a must be a whole number contained in R, n times.

n = whole number (reduction)

Crank bearing load/cam plate

$$F_{CB} = \frac{HP \times 63000}{4 \times rpm \times e}$$

F_{CB} = 3000 pounds in preliminary design

Resultant crank bearing load

$$P_{CB} = \sqrt{F_{CB}{}^2 + me\,\omega^{2*}}$$

*Dynamic load (Figure 3, Drawing 6144D47)

Where:

$m = \frac{\omega}{g}$ Mass of all pieces contained in a cam plate lamina

Cam Surface Hertz stress

$$Hertz_{max} = \left[.175 \times \frac{E \times (1/R + 1/r)}{L} P_{max}\right]^{1/2}$$

Hertz max should not exceed 324,000 psi

$$P_{max} = \left[P_{Drive\ Load} + D_{Dynamic\ Load}\right]_{max}$$

Reaction roller needle bearings

Weighted Equivalent Average $\Big\rangle$ $\text{Load} = F_R = \frac{\Delta X}{3xn}\left[y_0 + 4y_1 + 2y_2 + 4y_3 + \cdots y_k\right]$

Where:

y_0, y_1, y_2, etc. = reaction roller load ordinates at 0, 1, 2, 3, etc.

y_0, y_1, y_2, etc. $\rightarrow$ reflects the sum of the drive and dynamic loads per Drawing 6144D47. (Figures 1, 2 and 3).

Base bearing selection on weighted equivalent load, input speed and 1000 hours B-10 life. Re-examine bearing for maximum Hertz stress.

$$324{,}000 \quad H_{max} = \left[.175 \times E \times \frac{(1/R + 1/r)}{L} P_{max} \right]^{1/2}$$

<u>Reverter eccentric needle bearings</u>

Weighted Equivalent Average > $\text{Load} = F_E = \frac{\Delta X}{3xn} \left[\boxed{y_0} + 4y_1 + 2y_2 + 4y_3 + \text{---} \; \underline{y_k} \right]$

Where:

y_0, y_1, y_2, etc = reverter load ordinates at 0, 1, 2, 3, etc.

y_0, y_1, y_2, etc → reflects the sum of the drive loads transferred to the reverter per Drawing 6144D47, plus the reverter eccentric dynamic load.

$$\text{Total bearing load} = F_E + me\omega^2$$

Where:

$m = \frac{W}{g}$ = eccentric mass

e = input eccentricity

ω = input rpm

Base bearing selection on weighted equivalent load plus the unbalanced dynamic force, input speed and 1000 hours B-10 life.

Re-examine bearing for maximum Hertz stress,

Where:

$$324{,}000 \quad H_{max} = \left[.175 \times E \times \frac{(1/R + 1/r)}{L} P_{max} \right]^{1/2}$$

<u>General statement on bearing design</u> when re-examination shows H_{max} exceeds 324,000 psi, reduce H to values within surface endurance limits by increasing R, r and possibly L.

Design

Cycloid cam reducers were reviewed as single and multistaged units. The multistaged units were restricted primarily to double stages. Modular design investigations involved the paralleling of primary reducer units to achieve greater capacity.

The single stage unit is universally adaptable in all reasonable reduction ratios up to, and including 150:1, dependent on the power requirements, input speeds and design balance desired. For heavy power requirements at high reductions and input speeds, the componentry becomes excessively large to safely accommodate the transmission and dynamic loads. A threshold configuration has been established, and is referred to a basic 280 horsepower unit with a 6000 revolution per minute input. In a 29:1 reduction, this unit compares very favorably with traditional designs of only 1/3 the overall reduction and by weight, 30 percent heavier.

Extending the reduction to 150:1, a transmission of 280 horsepower with 30,000 rpm input necessitates staging. Recourse to a 10 x 15 or 12 x 12 reduction is an approach open to this requirement.

Jet engine capacities are augmented by adding stages without appreciably affecting frontal areas. A modification of this design attitude is predicated in the proposed modularization program for transmitting power in multiple levels of 280 horsepower (Drawings 6144R43 and 45, and 6144D48, 49 and 51). These basic units become featured building blocks, applicable to engines requiring transmissions capable of handling 280-2520 horsepower.

Employing these units in parallel requires gear couplers. As these modular drives are uniformly disposed, in a radial pattern, about the input and output centers, the gear couplers fore and aft can contribute some primary and tertiary reduction. This makes possible a lower intermediate, but high order reduction in the cycloid stage of 19:1. A small module envelope is achieved which predisposes the overall configuration to lesser dimensions. The basic 29:1 reduction is designed to accept an input speed of 6000 rpm, as compared with 4760 rpm for the 19:1.

To achieve output speeds of 200 revolutions per minute, the 29:1 units require no reduction in the power collector gears when operated in parallel. The 19:1 units fashioned into a similar grouping require a 1.26:1 reduction from the cycloid stage to the output gear. Thus a 24:1 speed drop or torque conversion is effected.

In multilevel 280 horsepower drives, where geometry discourages any augmentation of the cycloids, 29:1 straight thru reductions are proposed. The 19:1 units will be reserved for modularized drives where the couplings will be conveniently accommodated with very small reductions at the fore and aft ends. These departures make possible reductions of the order of 150:1 or less, with only a change in the front gearing.

Standardization by this recourse provides a spectrum of modularized high horsepower cycloid drives with the aforementioned 29:1 and 19:1 basic reduction units.

The 2500 horsepower transmission displayed schematically on Drawing 6144D48 is a singular application of the 29:1 unit used in a multiple array. Complexity is minimized because no additional reduction is needed. All gear couplings are locked in at 1:1. Contrariwise, the 19:1 unit requires added reduction gears fore and aft in addition to the 1:1 basic couplers.

The 29:1, 2500 horsepower configuration applied to a 150:1 overall reduction requires only front reduction gears with a 5:1 ratio, in addition to the distributors, but no reducers to accommodate the collector gears. The comparable 19:1 configuration commands a small girth dimension, but more complexity due to the inclusion of reduction gears along with the couplers.

The preference of one or the other is a function of the explicit application where a salient figure of merit will order the selection.

The 840 horsepower configuration (Drawing 6144R51) benefits more profitably from the adoption of the 19:1 cycloid reducers. The geometry lends itself to the dual function performed by the gear couplers doubling as reducers.

The cycloid reducers have limitless possibilities when properly applied. The study has centered about 19:1 and 29:1 units as building blocks due to their flexibility for incorporation into Helicopter and VTOL systems without exceeding existing configuration envelopes and promising weight reductions of 30 percent of that of traditional transmissions.

The foregoing designs are all endowed with an auto-balance feature which ensures their smooth and vibration-free action. The epicycloid cams, because of their gyrations, develop very high unbalanced dynamic loads. The cam's lack of fixed phase relationship with the input crank rotates the cam's center of gravity about that of the eccentric. The best corrective balance is achieved by separating the total cam into 3 laminae, establishing the thickness of the middle one as twice the width of the adjacents and finally phasing the middle plate 180 degrees out of registration with the other two. Thereby, not only is the dynamic load balanced, but also the drive load. The fore and aft crankshaft bearings are thus able to be minimized because they are load balanced and serve merely as spinners.

An unusual advantage of these drives despite their high reduction, is their complete freedom from backlash. If any exists, it is not perceptible. The 18:1 display model exhibited this remarkable action, and yet it was devoid of any bind.

Despite the emphasis placed on the units under consideration, the study does not imply their usefulness is limited to these modules. Rather, they were used as reasonable and well-considered examples to display the favorable characteristics of the cycloid reducer. Further is written about variations on this drive and its implications under PLANS FOR CONTINUING PERFORMANCE.

Specification for 280-Horsepower Transmission

I. INTRODUCTION

This specification sets forth the design objectives, configuration, design loading, test program and data requirements for a 280-horsepower helicopter transmission.

II. DESIGN OBJECTIVE

A. The transmission shall be designed to have an overhaul life of 1000 hours.

B. The transmission design target weight, including main rotor drive shall not exceed 112 pounds.

C. The rotor shaft outside diameter or contour shall be such as to accomodate test stand adaptors for U.S. Army test stand, drawing number AA-17A-0056, dated 19 May 1960.

D. Consideration shall be given to assuring a minimum noise level.

E. The transmission is an experimental model to prove out the cycloidal cam principle. It is not required to be airborne, but is designed to meet general helicopter specifications such as MIL-T-5955A and MIL-T-8679.

III. CONFIGURATION

A. The transmission shall incorporate a basic auto-balance cycloid cam reduction system.

B. Aircraft MIL lubrication oils shall be used. Pressure lubrication from an external oil supply shall be incorporated.

IV. DESIGN LOADING CONDITIONS

The transmission shall be designed for 280-horsepower with an input speed of 6000 rpm and an output speed of 315 rpm at the rotor mast. Transmission reduction ratio shall be 19 to 1.

A. Power Loading Conditions

1. Power Train

For bearing analyses, the following percentage operating time breakdown will be used:

Full Rated Power - 15%
90% Rated Power - 35%
75% Rated Power - 50%

Torsional oscillating load of ±25% shall be superimposed on the above loads.

A peak load of two (2) times full rated power shall be used as a strength criteria.

B. Flight Loads

The flight loads applied at the teetering pin of the rotor are as follows:

1. Maximum Load Condition

a. Vertical . . . . 7,960 pounds (Along rotor shaft centerline) (Upwards)

b. Horizontal . . . . . . 1,400 pounds (Perpendicular to rotor mast) (any direction)

2. Average Load Condition

a. Vertical . . . . . . 4,020 pounds (Upwards)

b. Horizontal . . . . . . 425 pounds (Forward)

The preceding flight loads are limit, or actually applied loads. To obtain ultimate loads for structural considerations, the above flight loads are to be multiplied by a factor of 1.5. Casting factors of 1.15 and 1.25 for limit and ultimate loads, respectively, shall be applied to all castings.

V. TEST PROGRAM

The transmissions shall be subject to experimental no-load testing and a 50-hour torque load test.

A. Test

1. The transmissions shall be tested as per Western Gear Corp. no-load test procedure.

2. The transmissions shall be torque load tested as per Western Gear Corp. 50-hour load test procedure. The loading method is based on the back-to-back principle.

The above tests shall be conducted by Western Gear Corp. at their facility. Western Gear Corp. shall conduct such other tests at their facility as they deem necessary to assure delivery of transmissions with correct assemblies.

B. Reports

Test results, including photographs and inspection of parts, shall be included in the test report.

VI. DATA REQUIREMENTS

Western Gear shall submit the following engineering data to USA TRECOM concurrent with the delivery of the experimental transmissions: -

A. One reproducible set of final drawings with two (2) copies shall be forwarded to the contracting officer.

B. Three (3) sets of completed final drawings shall be shipped with the transmissions.

I. INTRODUCTION

This specification sets forth the design objectives, configuration and design loading for an 840-horsepower helicopter transmission.

II. DESIGN OBJECTIVE

A. The transmission shall be designed to have an overhaul life of 1000 hours.

B. The transmission design target weight, including main rotor drive shall not exceed 336 pounds.

C. The rotor shaft outside diameter or contour shall be such as to accommodate test stand adaptors for U. S. Army test stand, drawing number AA-17A-0056, dated 19 May 1960.

D. Consideration shall be given to assuring a minimum noise level.

E. The transmission is an experimental model to prove out the modular concept of the cycloidal cam reducers. It is not required to be airborne, but is designed to meet general helicopter specifications such as MIL-T-5955A and MIL-T-8679.

III. CONFIGURATION

A. The transmission shall incorporate three (3) basic cycloidal cam reduction systems.

B. Aircraft MIL lubrication oils shall be used. Pressure lubrication from an external oil supply shall be incorporated.

IV. DESIGN LOADING CONDITIONS

The transmission shall be designed for 840-horsepower with an input speed of 6000 RPM and an output speed of 207 RPM at the rotor mast. Transmission reduction ratio shall be 29 to 1.

A. Power Loading Conditions

1. Power Train

For bearing analyses, the following percentage operating time breakdown will be used:

Full Rated Power - 15%
90% Rated Power - 35%
75% Rated Power - 50%

Torsional oscillating load of ± 25% shall be superimposed on the above loads.

A peak load of two (2) times full rated power shall be used as a strength criteria.

B. Flight Loads

The flight loads applied at the teetering pin of the rotor are as follows:

1. Maximum Load Condition

a. Vertical . . . . 17,500 pounds (upwards)
(along rotor shaft centerline)

b. Horizontal . . . . 3,080 pounds (any direction)
(perpendicular to rotor mast)

2. Average Load Condition

a. Vertical . . . . 8,750 pounds (upwards)

b. Horizontal . . . . 920 pounds (forward)

The preceding flight loads are limit or actually applied loads. To obtain ultimate loads for structural considerations, the above flight loads are to be multiplied by a factor of 1.5. Casting factors of 1.15 and 1.25 for limit and ultimate loads, respectively, shall be applied to all castings.

V. TEST PROGRAM

At a future time, a coordinated meeting will be held with TRECOM to establish a test program and procedure.

CYCLOIDAL CAM TRANSMISSION CONFIGURATIONS

The kinematic data and geometrical results tabulated in the accompanying Table 1 are compiled for ready reference. The Table is an abstract, published in Machine Design of February, 1947. The abstract is from an original paper by Allan H. Candee of the Gleason Gear Works of Rochester, New York, on the general principles of geometry and kinematics as applied to the design of rotating disc cams with flat followers having straight line motions. For any of the listed cam profiles, the follower motions resulting therefrom are fully described along with the kinematic features of a particular system's inversion. In each of the five cases, the motion has a simple mathematical description from which the related curves can be directly described and identified.

In the same sense, inversion of the epitrochoid reducer system changes the input-output ratio, and effects an output rotation of opposite or same-hand, referred to the input.

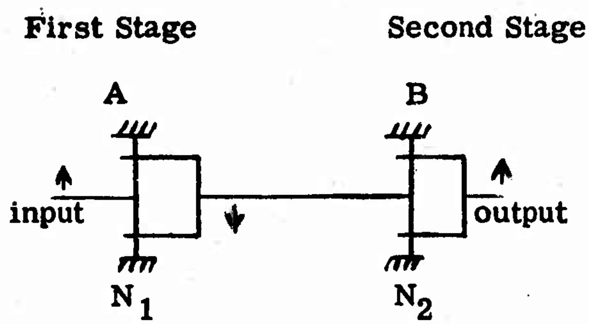
Cycloidal Cam Transmissions have four basic elements. These consist of eccentric, cam plate, reaction pins and reverter pins.

Transmissions can be designed using several combinations of elements. In the several arrangements, power input and power output elements differ. The possible combinations are listed below. See Item 28, Summary, for the other arrangements (Type E, F, G, H).

Type	Power Input To	Power Output From	Stationary Element	Output Ratio	Directions of Input and Output
A	Eccentric	Reverter Pins	Reaction Pins	$\frac{1}{N}$	Opposite
B	Eccentric	Reaction Pins	Reverter Pins	$\frac{1}{N+1}$	Similar
C	Reverter Pins	Reaction Pins	Eccentric	$\frac{N}{N+1}$	Similar
D	Reaction Pins	Reverter Pins	Eccentric	$\frac{N+1}{N}$	Similar

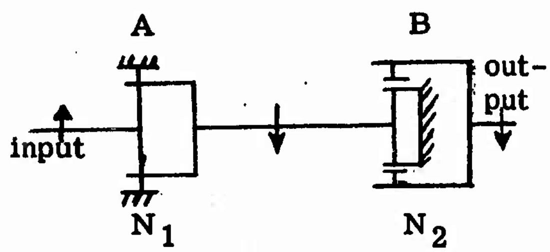
The foregoing considers only a single stage design. When considering a multi-stage design, various combinations are possible, each of which may have advantages or disadvantages for a specific application. The various possible arrangements to consider for a two stage design follow.

TWO STAGE COMBINATIONS



$$\text{Ratio} = \frac{1}{N_1 N_2}$$

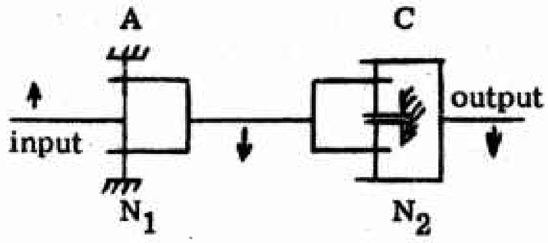
Power input rotates 1st stage eccentric
1st stage reaction pin cage stationary
1st stage reverter pin cage drives 2nd stage eccentric
2nd stage reaction pin cage stationary
2nd stage reverter pin cage drives the load



$$\text{Ratio} = \frac{1}{N_1(N_2+1)}$$

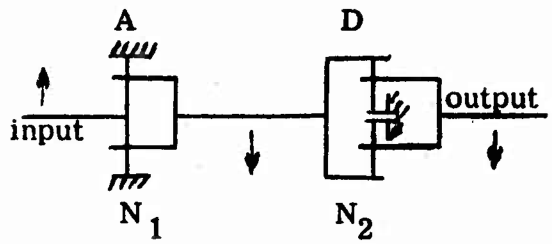
Power input rotates 1st stage eccentric
1st stage reaction pin cage stationary
1st stage reverter pin cage drives 2nd stage eccentric
2nd stage reverter pin cage stationary
2nd stage reaction pin cage drives the load

First Stage Second Stage



Ratio = $\frac{N_2}{N_1(N_2+1)}$

Power input rotates 1st stage eccentric
1st stage reaction pin cage stationary
1st stage reverter pin cage drives 2nd stage reverter pin cage
2nd stage eccentric stationary
2nd stage reaction pin cage drives the load



Ratio = $\frac{N_2 + 1}{N_1 N_2}$

Power input rotates 1st stage eccentric
1st stage reaction pin cage stationary
1st stage reverter pin cage drives 2nd stage reaction pin cage
2nd stage eccentric stationary
2nd stage reverter pin cage drives the load

Other possible combinations are:

B to A	C to A	D to A
B to B	C to B	D to B
B to C	C to C	D to C
B to D	C to D	D to D

Isometric drawings have been made of four of the basic single stage combinations (A, B, C, D) and one of a two stage (A to A). These serve to better show the action and point out some problems that may be encountered in future designs. These drawings are:

Figure	
4	Type A single stage
5	Type B single stage
6	Type C single stage
7	Type D single stage
8	Type A to Type A Two Stage

CAM LOBE STUDY

Figure 9 depicts a composite cam lobe study for various configurations. The different contours reflect cam designs considered for staged reductions. Applied to tandem units employing 1.00 inch diameter reaction rollers, it provides a ready comparison of compatibilities for the selection of areas available for reverter pins, crank bearings and cam lobe contours. The 1.00 inch diameter roller has been retained as a design parameter in most approaches to date.

LUBRICATION

Cycloidal Cam Transmissions present no new lubrication problems. Cam transmissions operated with a horizontal shaft axis could have lubrication provided by splash and/or directed oil flow.

Cam transmissions operated with a vertical shaft axis could have lubrication provided by directed oil flow.

Housings for either horizontal shaft or vertical shaft designs could have the required oil pressurizing pump incorporated and transmission driven to produce oil flow. An alternate scheme would be to provide a separate oil reservoir with an attendant pump separately driven. In such case, oil would be piped to the transmission for directed oil.

Standard aircraft oils are recommended, such as MIL-L-7808, MIL-L-6086, MIL-L-6081, MIL-O-6082, MIL-L-7870. These oils are preferred for aircraft transmissions because of their broad temperature operating range and lubrication transmission qualities. In particular, MIL-L-7808D has the broadest temperature range, good anti-foaming characteristics and fair gear and anti-friction bearing lubrication properties. MIL-L-7808D has a shelf-life of 5 years, pour point at -75° F and flash point at +400° F. Of the MIL oils listed, it is recommended that MIL-L-6086 be used for the transmission.

DESCRIPTION OF MODELS

The model was made to illustrate the Cycloidal Cam Reducing Principle. For model purposes, a simple, 3-lobe cam was used, as this facilitated hand fabrication and provided the fewest cam lobes to describe the action. In fact, this cam was cut out of plywood, using Drawing 6144E5 as a template.

In order to make the cam motion more easily observable, the output members, which in this case consist of 3 reverter pins and a spidered cage, were not assembled to the model.

The input, in the model, is represented by a disc upon which is painted a white dot. The output is represented by an arrow painted on the cam face.

The three (3) following photographs depict one complete revolution of the cam, in three positions:

1st Postition (Photographs L-9695 and L9697)

Input dot and output arrow line up in Photograph L-9695. The input revolves once counterclockwise, while the output arrow makes 1/3 revolution clockwise, as shown by Photograph L-9697.

2nd Position (Photographs L-9697 and L-9696)

Input dot again makes a complete counterclockwise revolution, while the output has now moved a total of 2/3 revolution in a clockwise direction, see Photograph L-9696.

3rd Position (Photograph L-9695)

A third revolution of the input dot in a counterclockwise direction brings the cam back to its original position as depicted by Photograph L-9695. Thus it will be seen that the cam has revolved once in a clockwise direction for three (3) revolutions of the input in a counterclockwise direction.

Model Results

Model No. 1

This was the simple, 3 lobe, wooden construction model made solely for demonstrating the cycloidal action of the system.

Bearing this in mind, this model served its purpose, but the operation of the working parts left a lot to be desired, mainly due to the unevenness of the cam plate, which was cut by a band saw blade, following a drawing print template.

Model No. 2 (Display Model Drawing 6144B27)

The study had proceeded far enough along for a serious attempt to be made to design and construct a more sophisticated model.

The release of drawings to Western Gear's Experimental Shop was completed by October 4, 1960, and fabrication of parts proceeded without snags arising. The cam plates were fly-cut with the tool shown in Photograph L-9917. The main components were ready for assembly by October 11th and fitted together extremely well. The rotating assembly was finally assembled to the outer case and base on October 17th.

The functioning of the finished assembly was pleasing. The smoothness and quietness of operation were immediately apparent, and backlash was undetectable. As this model is a miniaturized version of the 280 horsepower unit, these results are very encouraging.

The reduction ratio of this model is 18:1 and the design incorporates the auto-balanced, 3 cam plate principle where the cam plates are phased at 180 degrees.

The model is illustrated by Figure 10 and Photographs L-9996, 9997 and 9980.

Photographs taken during the manufacture of the model components are:

- L-9914 - Cam Cutting
- L-9915 - Finished Cams
- L-9916 - Cams Being Checked for "Phasing"
- L-9917 - The Fly-Cutting Tool for Cam Form.

Computer Research

The expressions for X_g and Y_g, parametric in $\emptyset$, cover the general case wherein the equidistant curve may be wholly above or below the base circle or have a crossover.

The development of the display model cams for an 18:1 reduction was special in that the cam surfaces did not cross over the base circle. A simplified expression for manual computation was developed for this purpose

where:

$$X_g = (R + a)\mathrm{Sin}\emptyset + b\,\mathrm{Sin}\left[\left(\frac{R+a}{a}\right)\emptyset\right] - \rho\left[\frac{(R+a)\mathrm{Sin}\emptyset + b\mathrm{Sin}\left(\frac{R+a}{a}\right)\emptyset - R\mathrm{Sin}\emptyset}{\sqrt{b^2+a^2-2ab\mathrm{Cos}(180^\circ - \frac{R}{a} \times \emptyset)}}\right]$$

$$Y_g = (R+a)\mathrm{Cos}\emptyset + b\mathrm{Cos}\left[\left(\frac{R+a}{a}\right)\emptyset\right] - \rho\left[\frac{(R+a)\mathrm{Cos}\emptyset + b\mathrm{Cos}\left(\frac{R+a}{a}\right)\emptyset - R\mathrm{Cos}\emptyset}{\sqrt{b^2+a^2-2ab\mathrm{Cos}(180^\circ - \frac{R}{a}\emptyset)}}\right]$$

The more general expression was programmed for the Univac 120 here at the Western Gear Corporation facility. The integrity of the two expressions was verified by the machine and manual computation. The results substantiated each other to the same degree of accuracy.

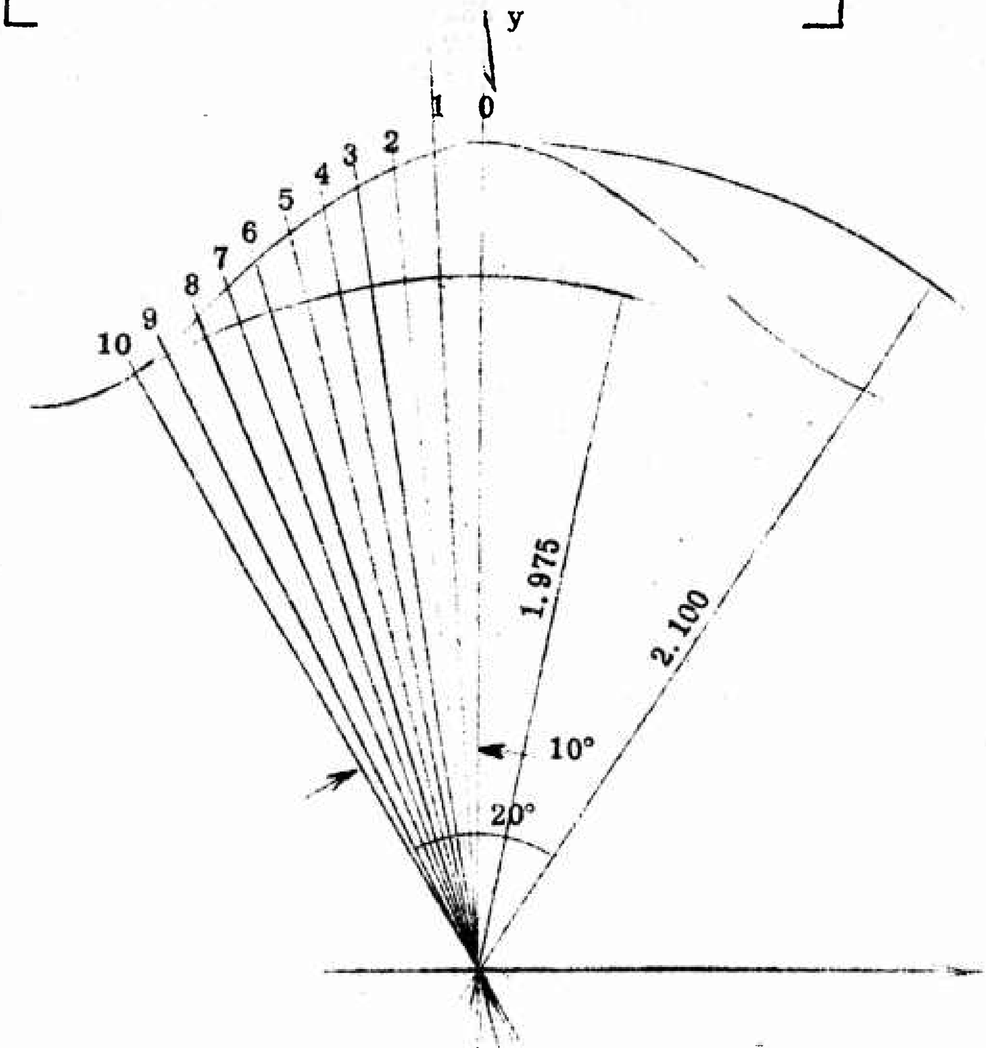
Figure 11 illustrates a typical programming card for the X_g and Y_g coordinates for any cycloid cam profile. The one displayed is a reproduction of the original card run to verify the 18:1 reduction data.

The cam cutter profile for the 18:1 reduction display model drive is depicted on:

Drawing Number 6144D38 (10x size)
Drawing Number 6144E39 (20x size)
Drawing Number 6144R40 (50x size)

Tabulated results from the evaluation of the various terms in the parametric equation for the Y_g ordinates over one cam lobe periphery follow. Similar tabulations are necessary for the X_g abscissas but are not included in this report, in the interest of brevity.

$$Y = \left[2.225 \cos \emptyset + .0625 \cos (19\emptyset) - .1875 \frac{2.225\cos\emptyset + .0625\cos(19\emptyset) - 2.1079\cos\emptyset}{\sqrt{.01762 - .01464\cos(180° - 18\emptyset)}} \right]$$



Sample of Data Processing Tabulations For Y_g Coordinates of Model Cycloid Cam Drive

ϕ	Cos ϕ	19 ϕ	Cos(19ϕ)	2. 225Cos ϕ	. 0625Cos19 ϕ
1	+. 99985	19	+. 94552	2. 22466	+. 059095
2	+. 99939	38	+. 78801	2. 22364	+. 049250
3	+. 99863	57	+. 54464	2. 22195	+. 034040
4	+. 99756	76	+. 24192	2. 21957	+. 015120
5	+. 99619	95	-. 08716	2. 21652	-. 005447
6	+. 99452	114	-. 40674	2. 21280	-. 025421
7	+. 99255	133	-. 68200	2.20842	-. 042625
8	+. 99027	152	-. 88295	2. 20335	-. 055184
9	+. 98769	171	-. 98769	2. 19761	-. 061730
10	+. 98481	190	-. 98481	2. 19120	-. 061550

Let

$$B = 2.225 \text{ Cos}\phi + .0625 \text{ Cos } 19\phi$$

ϕ	B	2. 1079 Cos ϕ	B-2. 1079Cos ϕ
1	2. 28375	2. 10758	. 17617
2	2. 27289	2. 10661	. 16628
3	2. 25599	2. 10501	. 15098
4	2. 23469	2. 10275	. 13194
5	2. 21108	2. 09986	. 11122
6	2. 18738	2. 09634	. 09104
7	2. 16580	2. 09219	. 07361
8	2. 14817	2. 08709	. 06108
9	2. 13588	2. 08195	. 05393
10	2. 12965	2. 07588	. 05377

ϕ	(18ϕ)	$(180° - 18\phi)$	$\cos(180° - 18\phi)$
1	18	162	-.95106
2	36	144	-.80902
3	54	126	-.58779
4	72	108	-.30902
5	90	90	0.-----
6	108	72	.30902
7	126	54	.58779
8	144	36	.80902
9	162	18	.95106
10	180	0	1.00000

ϕ	$0.01464\cos(180° - 18\phi)$	$.01762 - .01464\cos(180° - 18\phi)$
1	-.013923	.03154
2	-.011844	.02946
3	-.008605	.02622
4	-.004524	.02214
5	-.00----	.01762
6	.004524	.01310
7	.008605	.00902
8	.011844	.00578
9	.013923	.00370
10	.01464	.00298

Ø	Y	X	20X	20_y	Δy
1	2.098	.036	.720	41.960	3.040
2	2.091	.070	1.400	41.820	2.900
3	2.081	.101	2.020	41.620	2.700
4	2.069	.129	2.580	41.380	2.460
5	2.054	.154	3.080	41.080	2.160
6	2.039	.176	3.520	40.780	1.860
7	2.021	.198	3.960	40.420	1.500
8	1.998	.226	4.520	39.960	1.040
9	1.970	.271	5.420	39.400	.480
10	1.946	.343	6.860	38.920	-----

$\emptyset$	$\sqrt{.01762-.01464\cos(180^\circ-18\emptyset)}$	$\frac{B-2.1079\cos\emptyset}{f(18\emptyset)}$
1	.178	.1897
2	.172	.9667
3	.162	.9319
4	.149	.8855
5	.133	.8362
6	.115	.7916
7	.095	.7748
8	.076	.8536
9	.061	.8840
10	.055	.9776

$\emptyset$	$.1875 \sqrt{\frac{B-2.1079\cos\emptyset}{f(18\emptyset)}}$	Y_g
1	.1854	2.098
2	.1812	2.091
3	.1747	2.081
4	.1660	2.069
5	.1567	2.054
6	.1484	2.039
7	.1452	2.021
8	.1506	1.998
9	.1657	1.970
10	.1833	1.946

DESCRIPTION OF PRELIMINARY DESIGNS

Introduction

A basic Cycloidal Cam Transmission consists of an input shaft with an eccentric attached. Rotatably mounted on the eccentric is a multi-lobed cam plate having X number of lobes of modified epitrochoid profile.

Rotation of the input shaft causes the cam plate to describe an eccentric orbit. The orbit is a circular path of diameter equal to the difference in height between the crest of one lobe and the valley between 2 adjacent lobes.

An X plus 1 number of reaction shafts/rollers are mounted in a fixed housing or cage concentric with the input shaft axis. These shafts are so positioned that when a shaft is in the valley between two lobes another shaft is resting on the crest of a lobe 180 degrees removed, when an even number of rollers is involved. For an odd number two rollers straddle the centerline 180 degrees out of position with the roller riding a cam crest. When the input shaft is rotated the cam plate moves in the before-mentioned orbit and rotates in the path of interference created by the reaction shafts. Because there is one less lobe than reaction shafts each revolution of the eccentric rotates the cam plate an amount equal to one cam lobe. For example, if a 19 lobe cam were eccentrically rotated within a 20 shaft reaction cage it would require 19 revolutions of the eccentric to cause the cam plate to make 1 revolution; thus the ratio would be 19:1.

The cam plate being the reduced speed output member, requires that means be provided to revert the cycloidal motion of the cam plate to a member rotating on the common axis. This is accomplished by providing holes or races in the cam plate and locating these races a uniform radius from the eccentric bore. These races engage the common axis. The cam reverter races rotate in a cycloidal path around the reverter but are always making a bearing. In this manner the cycloidal motion of the cam plate is reverted to rotary motion of the reverter cage which is the final output.

The foregoing considers the use of one cam plate which results in unbalanced forces which are undesirable in a heavily loaded device. By the use of 2 cam plates and their eccentrics, all in phase, and disposed on each side of another cam plate and eccentric, dephased 180 degrees, dynamic balance of the forces is achieved. This balancing of forces permits a more compact and efficient design for a given load.

Considering the cam and fixed rollers, there are several important relationships which must be observed:

1. Cam Lobe (Epitrochoid) Generation, Drawing 6144E9

   R = Base Circle Radius

   r = Generating Circle Radius

   b = Tracing Point Radius

   n = Number of Cam Lobes

   S = Stroke

   $$r = \frac{R}{n} \qquad R = nr \qquad S = 2b$$

2. System, Drawing 6144E5

   $$\frac{n}{1} = \frac{R}{a} = \text{Ratio of System or Stage}$$

   e = eccentricity = b

   Number of fixed pins or rollers = n + 1

   Pitch diameter of fixed pins = 2(R + a)

   Pitch angle of fixed pins $= \frac{360}{n+1}$ Degrees

Drawing 6144B3 is schematic only, but depicts a two stage conception of a cycloidal cam reducer. It shows one approach to the problem of dynamic balancing by employing two cam plates per stage and phasing the plates 180 degrees apart. Where weight considerations permit, another balancing approach would be to employ three plates per stage, phased 180 degrees apart.

Drawing 6144E9 shows the profiles of a 30:1 cam for a single stage cycloidal reducer. It will be noted that the cam profiles are produced by a cutter whose center follows a path coincident with the closed epitrochoid curve traced out by the tracer point. As in normal cam practice, the diameter of the fixed pins or rollers must equal the diameter of the cutter used. Two sets of cam profiles are depicted on the drawing, both produced by the same basic generating data, but with different size cutters: namely, 1.0 inch diameter and 1.50 inches diameter. It will be noted that the two sets of profiles are quite different in shape.

By the cam profiles depicted in Drawing 6144E9, there is an optimum cutter size which will produce (for a given base circle radius) the optimum cam profile. This means there is also an optimum roller size for the same base circle radius. Programmed evaluation in this area will culminate in layouts and curves showing optimum roller diameters for variations of base circle radii.

<u>Drawing 6144R43</u>

The drawing shows the general layout of a Cycloidal Cam Transmission designed to the following specifications:

Input Horsepower	280
Input Speed	4760 rpm
Overall Ratio	19:1
Output Speed	250 rpm
Design Life	1000 hours

While this is not a finalized design, it is considered sufficiently correct to permit preliminary weight and manufacturing analyses to be made.

The modular design was evolved from an analysis of the forces involved, and from recommended bearing types and sizes suitable to meet the specified operating conditions.

A housing is not shown as the design would be dictated by mounting requirements and external loading. However, the weight of such a housing can be closely approximated from past experience.

Further study should result in design refinements with consequent weight reduction, reduction of manufacturing problems and savings in cost.

Modular type construction has also been considered for the design of transmissions in the larger horsepower ranges. It appears feasible to construct transmissions of greater power capabilities by grouping two or more basic units around a common input and output gear set. For example: 3 basic 280 horsepower type units, nested and gear coupled, could handle 840 horsepower input, (see Drawing 6144D51), provided the basic units were designed with suitable ratios to accommodate the speeds.

Drawings 6144E12, 6144E19, 6144R24

Drawing 6144E12 shows a typical 30:1 ratio cam system, which consists of one cam plate, 31 fixed rollers and 15 reverting rollers. This drawing was the basis for studying the fixed roller loads by the four-bar linkage method, and was used, also, for the setting up of cam lobe and reverter hole tolerances for manufacturing considerations.

During the investigation of crank and main bearing loads, it became evident that the small amount of eccentricity (.125 inch in the chosen system), would give rise to high tangential loads, and in order to reduce these and thus the sizes of the crank bearings required, a study was conducted of the effect of increased eccentricity on the cam lobes.

Drawing 6144E19 shows, at the left-hand side, the results obtained by making the eccentricity equal to the generating circle radius; that is, producing, in effect, a true epicycloid path for the cutting tool to follow. It will be noted that in the two cases depicted, using .50 and 1.0 inch diameter cutters, the cam profiles are destroyed by cutter interference, due to the existence of steep cusps in the epicycloid cutter path.

The right-hand side of the same drawing shows, that merely modifying the amount of eccentricity to be something slightly less than the generating circle radius - in this case a value of 80 percent of r = .20 inches, an epitrochoidal cutter path is again obtained, resulting in reasonable cam profiles for 3 sizes of cutters (.75, .875 and 1.00 inches diameter).

Information derived from the concurrent bearing load study indicated that for balance and load sharing reasons, a 3-plate cam was desirable for a 280 horsepower single stage version of a cycloidal cam reducer. Consideration of the phasing of 3 plates at 120 degrees shows that the following conditions would need to be met:

The number of fixed pins and reverting pins to be divisible by 6. Hence, as n = number of cam lobes and n + 1 = number of fixed pins, making n + 1 equal to 30, makes n equal to 29. Reverting holes used could be 6, 12, 18, 30, etc. in number.

Drawing 6144R24 is a preliminary layout of a 29:1 ratio, single stage 280 horsepower, 6000 rpm cycloidal cam reducer embodying the 3-plate cam design. The 120 degree phased cam plate design was subsequently abandoned due to the inherent imbalance.

Drawing 6144R45

The tentative design shown on Drawing 6144R45, represents a complete transmission unit less lubrication provision. It is contemplated that lubrication would be accomplished by oil mist, but this would be finalized in an engineered manufacturing design. The oil pressurizing pump, and associated mist producing sprays could be incorporated in the housing, or the pump and reservoir could conceivably be a separate system with oil piped to the spray nozzle within the housing. In such case an oil return pipe would return excess oil to reservoir.

As shown on the tentative design, all surfaces are engaged through anti-friction bearings to eliminate any sliding contact.

In addition to the 280 horsepower unit, the single stage cycloid cam drive investigations were expanded to include the 840, 1680 and 2520 horsepower units. Increasing reduction ratios commanding larger cam diameters require larger housing diameters. In turn, the dynamic loading of the reaction rollers, due to the gyrations of the cam plates, become prohibitive with increasing diameters and increasing speeds.

The drive elements need be over-designed to accommodate the excessive dynamic loads in addition to the drive loads. The bearings consequently become larger as do the peripheral speeds.

Single Stage Roller Cam Transmission

The size between the maximum diameters of the Single Stage Cycloid and Roller Cam configurations is very similar, see Figures 14 and 17. Despite this, the weight estimates of the Single Stage Roller Cam units are lower. In size, weight and over-all length, there is little on which to preference a selection between these two.

Figure 15 displays the weight and horsepower relationships with changing reduction ratios. Figure 16 reflects the weight and output torque characteristics for different reduction ratios. The linearity of the curves is sustained because circulating horsepower losses were not considered (ratios above 29:1). The circulating horsepower losses are generated mainly by the high dynamic loads which the reaction rollers must support. Figures 18, 19, 20 and 21 display the absolute values of the dynamic and drive loads with changing speed ratios at the different power levels.

Drawing 6144R63 illustrates a full-scale roller cam plate arrangement for an 840 horsepower, 29:1 reduction unit. Figure 22 illustrates a complete assembly founded on a cam plate arrangement similar to that of the referenced drawing.

Drawing 6144D51

The transmission was designed to the following specifications:

Input Horsepower	840
Input Speed	6000 rpm
Output Speed	207 rpm
Reduction Ratio	29:1

The complete package consists of three nested cycloidal cam modules coupled by gear trains to input and output, designed so that each unit will take 33-1/3 percent of the load. This is accomplished by providing elastic torque shafts for the gear trains coupling the 3 identical modules. These torque shafts would be preloaded in assembly. The modules incorporated are shown on Drawing 6144D51 and described in further detail under Drawing 6144R43.

Drawing 6144R22

This was a preliminary design produced for analysis. It was one of the first designs attempted to produce an 800 horsepower unit having a reduction ratio of 30:1. In this design, all cam units are grouped around the common axis of input and output shafts.

An analysis indicated that this design would be too heavy and bulky. Therefore, work on this approach was discontinued. Design information obtained was helpful in subsequent designs of 800 horsepower transmissions.

Drawing 6144D50

Multi-stage Cycloidal Cam Transmissions can also be designed by nesting two or more stages. Again these stages may be any of the before-mentioned four basic configurations, mentioned in the SUMMARY.

A two-stage, nested transmission could consist of a basic Type A nested within a Type A. In such a design the action would be as follows:

Power input rotates 1st stage eccentric.
1st stage reaction pin cage stationary.
1st stage reverter pin cage rotates 2nd stage eccentric, which surrounds the reverter pin cage and connected to it.
2nd stage reaction pin cage stationary.
2nd stage reverter pin cage rotates the load.

The arrangement of the two stage configuration is shown on Figure 23. This illustration explains how the two stages telescope together.

Drawing 6144D48

This design consists of identical modules grouped about a common axis.

As shown on the drawing, an input shaft has an attached gear engaging with 4 driven gears. An extension of the input shaft is also attached to the power input end of a single module operating on the input shaft axis. The four referenced driven gears are attached to four shafts that go through the axis of eight (8) modules; 2 on each shaft. These shafts are connected to the input end of each module. The outputs of each pair of modules are connected together by a tubular shaft operating concentric to the axis of the shafts. Each of these tubular shafts has a gear attached which comprises the driving gears of a collector gear set. These driving gears drive a driven gear attached to the output shaft.

The foregoing method of grouping and coupling results in a diversion of input power to all nine modules whereby each module supplies a proportionate share of output power to a common shaft.

Also shown on the drawing are alternate arrangements of gear sets to give greater overall ratios.

The described unit is comparatively compact considering the overall ratio and power input. However, due to the nested arrangement, the lubrication system would be more complex than in other concepts that have been considered.

Drawing 6144D49

Basically, this concept consists of 3 groups of three 280 horsepower transmission modules (total 9) gear-coupled to provide a common power input and output, and disposed around a common axis.

As shown, a central driving gear attached to the input shaft engages with three driven gears. Each of these driven gears is mounted on a shaft extending through three (3) identical cam transmission modules. These shafts are coupled to the input of each module. The outputs of each of the modules terminates in a gear. These gears engage driven gears attached to a common output shaft. To ensure an equal distribution of load to each module, the coupling shafts are designed to be torsion members, and the entire unit would be assembled with a designed pre-load on the torsion shafts.

The modules in this design are 19:1 ratio. However, the final ratio would be determined by the ratio of input and output gearing. With this arrangement the overall ratio could be less or greater than the module ratio provided that the input speed to the modules does not exceed the designed maximum.

The design shown is compact and elastic in application, due to the great ratio possibilities.

Differential Cam Design

Investigation into the feasibility of a differential cam design disclosed the possibility of producing a single stage, double reduction cam transmission, using the basic 3 cam-plate, auto-balanced concept. Higher ratios are possible with such arrangements.

This is accomplished in a basic 3 cam-plate concept by introducing the input into one plate and roller assembly of a 3-plate-cam group operating as a unit; and taking the output from the other two roller assembly units, which are mechanically separated from the input roller assembly.

In operation, the input rotates the input eccentric, thus rotating the input cam plate at a designed reduced ratio because of the action between the cam plate and the rollers. The input cam is coupled to the two output cams by reverter pins, and the cam plates are rotated in an eccentric path by 2 additional eccentrics coupled to the input shaft, and rotating at input speed. The action of the two output cams rotating in an eccentric orbit against the output roller assembly, causes rotation of the output roller assembly at a selected reduced ratio from that of the input cam plate.

Drawing 6144R64 is a kinematic study of a 5 X 3 Differential Cam Reducer. The 5-lobed cam precedes the 3-lobed cam on the drive axis with respect to the input-output sense. In action, the 5-lobe cam effects a 5:1 reduction of the input speed of the eccentric shaft at the reverter cage. The reverter cage output rotation of the 5-lobe cam is opposite to the input. Through this reversion, the 3-lobe cam assumes the burden of two inputs. One is the eccentric shaft input, and the other is the reverter cage input at 1/5 the eccentric shaft speed. These inputs to the 3-lobe cam rotate in opposite directions. An inversion of the cam kinematics employing a fixed reverter cage, an eccentric shaft input, and a reaction roller cage output, effects a reduction of (N+1) or 4:1. A supplementary inversion practiced in the differential cam arrangement secures the eccentric shaft, drives the reverter cage, and establishes the reaction roller cage output reduction as $\left(\frac{N}{N+1}\right)$ or 3/4. As noted on the drawing, the overall reduction then becomes $N\left(\frac{N_2+1}{N_1-N_2}\right)$ or $\frac{5 \times 4}{3-5} = -10$, signifying a 10:1 reduction with output rotation opposite to input. Thus, the resultant action is a combination of the two inversions occurring simultaneously. The two outputs always have opposite directions. The magnitudes and signs of these determine whether the final output is opposite to the input, or the same. The analytical expression establishes the fact in the following manner as demonstrated earlier:

When $N_2 < N_1$ input and output rotate in the same direction.
When $N_2 > N_1$ input and output rotate in opposite directions.

Figure 24 illustrates a practical arrangement for a transmission providing 3 cam plates for each reduction phase. Figures 25 and 26 are graphs of weight and housing diameters, plotted against reduction ratios for the different power level units of the study.

The increase of weight with decreasing reduction ratios is contrary to the characteristics generally associated with transmissions. This trend firmly dictates a place for their application in the higher ratios and the higher horsepower spectra of the study. The lower powered units perform more tradionally with respect to weight and ratios. Though all units of the differential type involved in the study are apparently heavier, when compared with other foregoing concepts, more promising weight/horsepower ratios are foreseen for units below 280 horsepower.

Load levels low enough to be serviced by a total of only 3-cam plates for both phases, instead of 6, will enable the differential cam concept to match weight/horsepower ratios with the best so far ascertained.

Maximum frontal area diameters plotted against ratio for the differential cams are smaller than those for the single stage concepts, but not as small as those of the modular arrangements.

ENVELOPE SIZE

The preliminary modular cycloid cam drive configurations were restricted to frontal area diameters not to exceed those of the engines. The two basic modules employing a 19:1 and a 29:1 reduction, have maximum outside diameters of 15.2 inches and 16.6 inches, respectively. These units are both of 280 horsepower capacity. Their frontal area diameters fall within the maximum offset frontal area dimension of the Allison T63, 250 horsepower turbo-shaft engine.

The 840 horsepower transmission designs are 24 inches maximum in diameter, which is less than that of the Lycoming T53 (24.25 inches diameter). In like manner, the 2520 horsepower units maximum frontal area diameter is 29.5 inches, which compares with General Electric's T64, 30.4 inches offset dimension.

Figure 27,28,29 and 30 are turbojet engine profiles which influenced the design trend phase of the study. Tables 6 and 7 enumerate the jet engines researched for accessibility to high and low speed power takeoffs, for future implementation with cycloidal cam transmissions. Various charts have been plotted with displacement and frontal area vs horsepower.

CALCULATIONS

The calculations developed for the transmissions are applied in a very basic manner, minus all refinements. In the preliminary load-stress analysis which follows, the sub-headings denote the study area. The transmission under review has an eccentricity or crank radius of .200, and with an efficiency (to be discussed later) of 90 percent.

Design Requirements

Capacity - 280 horsepower

Input rpm = 6000

Output rpm = 200

B-10 Life = 1000 hours

Load Schedule

Full rated power - 15%

90 percent rated power = 35%

75 percent rated power = 50%

Weighted Average Horsepower

$$\text{Horsepower weighted} = \frac{100}{\frac{15}{HP} + \frac{35}{.9HP} + \frac{50}{.75HP}}$$

$$= 232 \text{ horsepower}$$

Bearing (B-10 life) 1000 hours based on 232 horsepower

4-Plate Cam Design - Auto Balance

2 center plates in phase

2 end plates out of phase

Input Torque

$$T = \frac{HP \times 63000}{rpm}$$

$$= \frac{232 \times 63000}{6000}$$

= 2, 440 pound inches

T/cam = 610 pounds

$$F/cam = \frac{T/Cam}{e}$$

$$= \frac{610}{.20}$$

= 3050 pounds
(radial bearing load/cam)

Output Torque

$$T = \frac{232 \times 63000}{200}$$

= 73000 pound inches

$$T/Cam/Pin = \frac{T/4}{n}$$

Where:

T/4 = Torque/Cam plate

n = Number of reverter pins

$$T/Cam/Pin - \frac{73000}{4 \times 12}$$

= 1530 pound inches

$$F/Pin = \frac{T}{L}$$

Where:

L = Reverter pin radial distance

$$F/Pin = \frac{1530}{5.5} = 277 \text{ pounds}$$

Dynamic Load

$$F = \frac{Mv^2}{e} = \frac{Wv^2}{ge} = \frac{Wew^2}{g} = \frac{We}{g}\left(\frac{2\pi rpm}{60}\right)^2$$

Where:

W = weight of whole cam plate lamina

g = gravity acceleration

e = eccentricity

From Drawing 6144D47 (Figure 3)

$$P\gamma_i \cong P_n \mathrm{Cos}^{3/2} \gamma_i$$

$$\Sigma P\gamma_i = P_n \sum_{k=1}^{n} \mathrm{Cos}^{3/2} \gamma_i = F$$

$\gamma_1 = 81.5°$ $\gamma_2 = 68.5°$ $\gamma_3 = 53.0°$ $\gamma_4 = 33.5°$ $\gamma_5 = 12.0°$

$Cos^{3/2}\gamma_1 = (.1478)^{3/2} = .057$

$Cos^{3/2}\gamma_2 = (.3665)^{3/2} = .221$

$Cos^{3/2}\gamma_3 = (.6018)^{3/2} = .470$

$Cos^{3/2}\gamma_4 = (.8387)^{3/2} = .765$

$Cos^{3/2}\gamma_5 = (.9782)^{3/2} = .960$

$F = P_n 2(.057 + .221 + .470 + .765 + .960) = 2000 = P_n \times 4.746$

$P_n = \frac{2000}{4.746} = 422$ pounds

$P\gamma_1 = P_n Cos^{3/2}\ 81.5^\circ = 422 \times .057 = 24.0$ pounds

$P\gamma_2 = P_n Cos^{3/2}\ 68.5^\circ = 422 \times .221 = 93.5$ pounds

$P\gamma_3 = P_n Cos^{3/2}\ 53.0^\circ = 422 \times .470 = 198.0$ pounds

$P\gamma_4 = P_n Cos^{3/2}\ 33.5^\circ = 422 \times .765 = 323.0$ pounds

$P\gamma_5 = P_n Cos^{3/2}\ 12.0^\circ = 422 \times .960 = 404.0$ pounds

Combining the drive and dynamic loads on the affected rollers, the new load values become:

$P_{12} /277 = .087$

$P_{13} /277 = 1.775$

$P_{14} /277 = 1.262$

$P_{15} /277 = 3.540$

$P_{16} /277 = 2.547$

Pin No.	Load P/277	Simpsons Rule Terms			
1	3.500	y_0	3.500		
2	1.500	$4y_1$	6.000	15.375	
3	2.937	$2y_2$	5.875		
4	1.000	$4y_3$	4.000		
5	1.750	$2y_4$	3.500	11.250	34.475
6	.937	$4y_5$	3.750		
7	1.687	$2y_6$	3.375		
8	.437	$4y_7$	1.750	7.750	
9	1.312	$2y_8$	2.625		
10	.312	$4y_9$	1.250		
11	1.312	$2y_{10}$	2.625	4.223	
12	.087	$4y_{11}$	.348		
13	1.775	$2y_{12}$	3.550		
14	1.262	$4y_{13}$	5.048	15.678	49.343
15	3.540	$2y_{14}$	7.080		
16	2.547	$4y_{15}$	10.188		
17	2.547	$2y_{16}$	5.094	29.442	
18	3.540	$4y_{17}$	14.160		
19	1.262	$2y_{18}$	2.524		
20	1.775	$4y_{19}$	7.100	9.789	
21	.087	$2y_{20}$	.174		
22	1.140	$4y_{21}$	4.562		
23	.250	$2y_{22}$	.500	11.312	34.226
24	1.562	$4y_{23}$	6.250		
25	.375	$2y_{24}$	.750		
26	1.812	$4y_{25}$	7.250	13.125	
27	.625	$2y_{26}$	5.125		
28	2.562	$4y_{27}$	10.250		
29	.750	$2y_{28}$	1.500	17.125	
30	5.375	y_{29}	5.375		

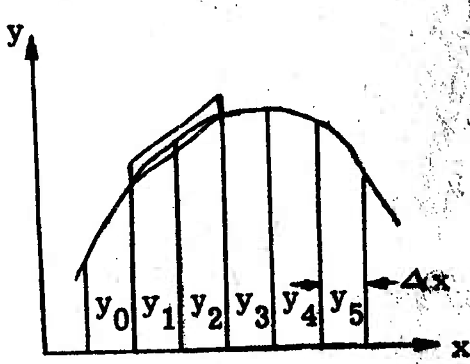
TOTAL 135.169

Stress

$$P = \frac{9.75}{386} \times .20 \times 4\pi^2 \times \left(\frac{6000}{60}\right)^2$$

= 2000 pounds

See pages 65, 66 and 67.



$$\text{Area} = \frac{\Delta x}{3}(y_0 + 4y_1 + 2y_2 + 4y_3 + \cdots y_n)$$

Where:

$\Delta x = 1$

Area = 1/3 x 135.169

= 45.056

$$\text{Effective (y)} = \frac{45.056}{30} = 1.50$$

Effective bearing load = 1.50 x 277
= 415 pounds

$$\text{Hertz}_{max} = \left[.175 \times P \times E \times \frac{(1/R + 1/r)}{L}\right]^{1/2}$$

Where:

R = Radius of curvature of cam lobe
r = Radius of curvature of reaction pin roller
P = Max contact load

$$H_m = \left[.175 \times 1490 \times \frac{3 \times 10^7 (1/.437 + 1/.500)}{.562}\right]^{1/2}$$

= 147,000 psi

Reaction Roller (Full complement needle)

Inner Race diameter = .625
Needle Roller diameter = .078
Number of rollers = 28
Average P = 39.2 pounds

$$P_{max} = 5 \times 39.2$$

$$= 195 \text{ pounds}$$

$$H_{max} = \left[.175 \times 195 \times 3 \times 10^7 \frac{(1/.312 + 1/.039)}{.562} \right]^{1/2}$$

$$= 229,000 \text{ psi}$$

<u>Reaction Roller Life</u>

$$W_{BDC} = 10,600 \times P.D. \times L$$

$$= 10,600 \times .703 \times .562$$

$$= 4200 \text{ pounds}$$

Where:

P. D. = .625 + .078

L = .562

Capacity @ 6000 rpm - 1000 hours (B-10)

$$C = \frac{W_{BDC}}{S.F. \times L.F.}$$

$$= \frac{4200}{6.0 \times 1.19}$$

= 590 pounds (Required 415 pounds)

<u>Reverter Pin Eccentrics</u>

$$\left.\begin{array}{l}\text{Reverter eccentric}\\ \text{maximum load}\end{array}\right\} \begin{array}{l} = 4.687 \times 277 \\ = 1295 \text{ pounds}\end{array}$$

$$\left.\begin{array}{l}\text{Unbalanced ecc.}\\ \text{dynamic load}\end{array}\right\} = mew^2$$

$$= \frac{1.05}{386} \times .200 \times \left(\frac{2\pi \times 6000}{60}\right)^2$$

$$= 216 \text{ pounds}$$

Where:

Weight = 1.05 pound

Gravity = 386 in/sec^2

$$\left.\begin{array}{l}\text{Total max.}\\ \text{reverter pin}\\ \text{load}\end{array}\right\} = P_{\text{Driveload}} + P_{\text{Dynamic load}}$$

$$= 1295 + 216$$

$$= 1511 \text{ pounds}$$

The reverter and reaction pin bearing loads are respectively 1511 pounds and 1490 pounds.

Reaction pin bearing is safe in Hertz and overcapacity in life. Reaction pin and reverter pin bearings are of approximately the same magnitude. Hence the reverter pin bearing elements are adequate.

Input Shaft (Basic)

Inside diameter = 1.60

Outside diameter = 2.08

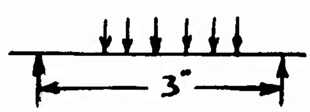
Torsion Stress

$$S_s = \frac{Txc}{I_p}$$

$$= \frac{2440 \times 1.04}{.098(2.08^4 - 1.60^4)}$$

$$= 2140 \text{ pounds per square inch}$$

Beam stress



Assume

$$\frac{PL}{8Z} < S_T < \frac{PL}{4Z}$$

$$= \frac{PL}{6Z} \text{ (approximately)}$$

Where:

$P = \frac{4 \times 3050}{.2} = 61000$ pounds (Assume full load sustained between outer cam bearings
P = Uniform Load)

$L = 3.0$ inch

$$S_T = \frac{PL}{6 \times .098 \left(\frac{D^4 - d^4}{D}\right)}$$

$$= \frac{61000 \times 3.00}{6 \times .098 \left(\frac{2.08^4 - 1.60^4}{2.08}\right)}$$

$S_T = 5300$ pounds per square inch

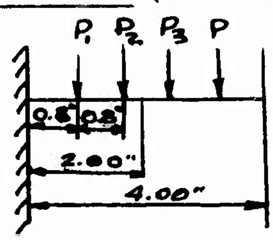
Combined Stress

$$S_{CS} = \frac{1}{2} S_T \pm \sqrt{S_S^{\,2} + \frac{1}{2} S_T^{\,2}}$$

$$= \frac{1}{2} \times 5300 \pm \sqrt{2140^2 + \left(\frac{1}{2} \times 5300\right)^2}$$

$= 6050$ pounds per square inch

Reaction Pin (Beam stress)



$$EI\frac{d^2y}{dx^2} = -M_1 + \frac{3}{2}Px$$

$$EI\frac{dy}{dx} = -Mx + \frac{3}{2}\frac{Px^2}{2} = -Mx + \frac{3}{4}Px^2 + C$$

When

$x = 0$
$\frac{dy}{dx} = 0$
$c = 0$

and

$x = L/2$

$\frac{dy}{dx} = 0$

$$\therefore 0 = -\frac{ML}{2} + \frac{3}{4}\frac{PL^2}{4} = -\frac{ML}{2} + \frac{3}{16}PL^2$$

Maximum positive bending moment occurs at center where vertical shear is zero $(v = 0)$

$$\frac{ML}{2} = \frac{3}{16}PL^2$$

$$M = \frac{3}{8}PL$$

$$S_T = \frac{Mc}{I} = \frac{M}{Z} = \frac{M}{.098\frac{(D^4-d^4)}{D}} = \frac{\frac{3}{8}PL}{.098\frac{(D^4-d^4)}{D}}$$

$$= \frac{.375 \times 1500 \times 2.00}{.098\frac{(.625^4-.375^4)}{.625}}$$

$= 50,000$ pounds per square inch

Reverter Pins (Beam Stress)

Loading is very much the same as for reaction pins. Since their physical dimensions are practically identical they, too, are safe.

Reaction Rollers (Outer Race)

Compressive stress (Hoop) not to exceed 50,000 pounds per square inch. (Recommendation - Torrington Needle Bearing Co.)

Inner race O. D. = .625
Roller diameter = .078
Outer race

Inside diameter	=	.781
Outside diameter	=	1.000
t (thickness)	=	.109

$$S_c = \frac{P}{2x\ t\ x\ L}$$

$$= \frac{1500}{2\ x\ .109\ x\ .562}$$

= 12,200 pounds per square inch (very safe)

The sensitive areas of the cycloid drive covered in the foregoing analysis display no excessive loading conditions. Small sectional increases in critical components can conceivably extend a unit's capacity.

Efficiency

The efficiency of cycloidal cam transmissions appears to be higher than that of comparable gear units. The sliding, interrupted meshing and dynamic loading of conventional gearing dissipates 1 to 2 percent of the input energy per mesh. The pure rolling action of the cycloid cam occasions no sliding losses. All the contact surfaces are antifriction bearings. Each bearing contact has a maximum rolling friction coefficient of .004 and at best of .002. Even with a great multiplicity of bearings, the friction energy dissipation is relatively low. At the present time, the following theoretical efficiency analysis has not been verified with any back-up test data.

The expression for the friction energy loss in bearings is:

$$HP_{\Delta E} = \pi \times \frac{PD \times rpm \times F \times C_F}{33000}$$

Where:

Δ_E = Friction energy

P.D. = Roller pitch diameter

F = Radial bearing load

C_F = Rolling Friction Coefficiency (.004)

Therefore, the efficiency of the pure cycloid reducer is:

$$\text{Efficiency (\%)} = \frac{HP_{In} - HP_{\Delta E}}{HP_{Fn}}$$

An analysis follows of the losses occasioned in a proposed 280-horsepower cycloid transmission operating at 6000 rpm input, and 200 rpm output. The distribution of losses are developed in the following manner:

Reverter and Reaction Pin Needle Bearings

$$HP_{\Delta E} = \frac{\pi \times D \times rpm \times F \times C_F \times n}{33000}$$

Where:

C_F (Bearing friction coefficient) = .004

F (Bearing radial load) = .415 pound

D (Bearing pitch diameter) = .781/12

Input speed = 6000 rpm

n (Number of same bearings) = 168

$$\text{Horsepower} = \frac{\pi \times .781/12 \times 6000 \times 415 \times .004 \times 168}{33000} = 10.35$$

Reverter Eccentric Bearings

$$HP_{\Delta E} = \frac{\pi \times 1.812/12 \times 6000 \times 592 \times .004 \times 48}{33000} = 9.75$$

Where:

F = 592

D = 1.812/12

n = 48

C_F = (Bearing friction coefficient) = .004

Input speed - 6000 rpm

Center Crank Bearing

$$HP_{\Delta E} = \frac{\pi \times 4.00 \times 6000 \times 6100 \times .002}{12 \times 33000} = 2.33$$

End Crank Bearing

$$HP_{\Delta E} = \frac{3}{4} \times 2.33 = 1.74$$

Bearing losses $\quad$ Sum = 10.35 + 9.75 + 2.33 + 1.74 = 24.2 horsepower

$$\text{Efficiency} = \frac{(232.0-24.2)}{232} \times 100 = 89.5 \text{ percent}$$

The losses in a 19:1 reduction might reasonably be expected to not exceed 75 percent of the losses occasioned in the 29:1 reduction. Thus, an efficiency loss of 7.5 percent might be assumed for the 19:1 reducer.

The 840 horsepower configuration as represented on Drawing 6144D51, employs three gear-coupled 19:1 units. Assuming a 2 percent energy loss to cover all contingencies at each gear mesh, the horsepower loss/power path reflected in the overall 30:1 reduction is:

$$HP_{\Delta E} = 280\ (2\% + 7.5\% + 2\%) \times 3$$

$$\text{Efficiency} = \frac{(840-96.3)}{840} \times 100 = 88.5\%$$

The same drive in a 150:1 model reducer might be expected to suffer an additional 2% power loss and reflect a depressed efficiency of 86.5 percent.

The 2520 horsepower, 30:1 module, with cumulative losses expressed by:

$$HP_{\Delta E} = (2\% + 10.5\% + 2\% + 10.5\%)\ 1120 + (2\% + 10.5\%)\ 280 = 315$$

Nevertheless exhibits,

$$\text{Efficiency} = \frac{(2520-315)}{2520} \times 100 = 87.5\%$$

Another stage of reduction added to increase the range of the 2520 horsepower unit to 150:1 burdens the drive with yet another 2.5% loss, and swells the drop to:

$$\Delta HP_E = 378 \text{ horsepower}$$

Or a figure for:

$$\text{Efficiency} = \frac{(2520-378)}{2520} \times 100 = 85\%$$

In like manner, the 19:1 module in a 2520 horsepower assembly develops an energy loss of:

$$HP_{\Delta E} = (2\% + 7.5\% + 2\% + 7.5\%)\ 2240 + (4\% + 7.5\%)\ 280 = 458$$

$$\text{Efficiency} = \frac{(2520-458) \times 100}{81.7} = 81.7$$

In an 840 horsepower array of the 19:1 modules, an expected horsepower drop should not exceed 11.5 or a transmission efficiency of 88.5 percent.

The foregoing plus some extensions of these results are tabulated for convenience.

EFFICIENCY (Percent)					
MODULES					
19:1 MODULE				30:1 MODULE	
OVERALL REDUCTION					
HP	19:1	30:1	150:1	30:1	150:1
280	92.5	----	----	89.5	----
840	88.5	88.5	86.5	88.5	83.5
2520	81.7	82.0	80.0	87.5	85.0

Weight

The estimated weight/horsepower ratio has been verified by the weight analyses of the preliminary designs. Table Number 8 is a breakdown of the 19:1 and 30:1, 280 horsepower units by major components. The 19:1 analysis is extended to the 840 horsepower unit which is a composite assembly of three integrated 19:1, 280 horsepower units.

The schematics, drawings 6144D48, 49 and 51, illustrate different configurations. Drawing 6144R43 and 45 disclose the design features of the modules on which the basic weight breakdowns were established.

The weight/horsepower ratio generated from the preliminary designs, varies from .3-.4 pounds/horsepower. The ratio rises with the higher reduction ratios. A comparison with contemporary transmissions, offering reduction ratios of only 8:1 to 11:1, is very favorable to the cycloid cam configurations. The weight/horsepower ratios of contemporary transmissions, dry weight, range from .33-.506 pounds/horsepower.

The presentations in Table 8 and 9 display the results of the analyses and help to emphasize the comparisons.

Transmission Dry Weight Study

Table 9 has been compiled from information furnished by the Directorate of Engineering, U.S. Army Transportation Materiel Command, St. Louis, and TRECOM, Fort Eustis, Virginia.

The information in this table was used as a standard for comparing the estimated weights of cycloidal cam transmissions with conventional transmissions.

Life

Reliability was established on a B-10 life of 1000 hours for all bearings. The most severe Hertz stress (280 horsepower module) amounted to 229,000 pounds per square inch, for a life of 125 x 10^6 stress cycles, or 343 hours. The use of vacuum melt Consutrode 8620 will extend the cyclic life to 400 x 10^6 stress cycles, or 1,111 hours (see figure 23).

Thus, recourse to more resistant materials extends the cyclic life of such vital componentry as the bearing elements.

TESTING

No Load Test Procedure

1. Weigh the module and photograph the first unit, also photograph the no-load test stand.

2. Examine the module carefully to make sure that it conforms to engineering requirements with respect to workmanship, finish and marking. Make sure the unit has a nameplate properly installed and completely filled out.

3. Mount the module on a suitable test stand in the normal mounting position. Mast should be up with oil sump down. (INPUT PAD).

4. The module requires an external lubrication system. Use MIL-L-6086, lightweight, as the lubricant.

5. Connect the three (3) inlet oil lines to the module. Connect the two drain lines to the module. Make provisions to measure the return oil. It may be necessary to have a scavenge pump, but try gravity return as the first approach.

6. At time of assembly of the module, omit the mast and its bearings and seals, as well as the output housing cover, in order to view the lubrication system of the cycloidal parts.

7. Circulate room temperature oil through the module with an oil pressure of 5 psi. Observe the oil circulation of the various components; such as:

   7.1 Crankshaft and bearings.
   7.2 Cam Plate reverter pins and bearings.
   7.3 Cam Plate fix pin rollers.
   7.4 Miscellaneous parts.

8. If there is any evidence that the lubrication system is incorrect, see layout drawing, then re-check the assembly and make necessary corrections and drawing checks.

9. Re-check the lubrication system for a nominal flow of 5 pounds per minute and record the oil pressure. (.7 gallons per minute).

10. Connect a suitable input power source to the input pad, preferably an hydraulic motor.

11. Record the following at five (5) minute intervals during no load operation:

    11.1 Output speed (RPM).
    11.2 Input speed (RPM).
    11.3 Oil in temperature (° F).
    11.4 Nameplate temperature (° F).
    11.5 Oil out temperature (° F).
    11.6 Ambient air temperature (° F).
    11.7 Noise level and/or vibration (Db).

12. No load test

Operate the module according to the following table. Bring the module up to indicated speeds gradually.

INPUT SPEED RPM	OPERATING TIME MINUTES
10	5
50	5
100	5
150	5
200	5
300	5
500	5
1000	5
1500	10
2000	10
3000	10
4000	10
5000	10
6000	10 minimum, operate until temperature stabilization

13. Measure the backlash of the module by locking the output and applying 10 inch-pounds at the input pad. Record in terms of degrees of arc.

14. Measure the windup of the module by locking the output and applying 2800 inch-pounds of torque. Record in terms of degrees of arc.

15. The module shall be rejected for any one of the following reasons:

15.1 Unit does not conform to paragraph 2.
15.2 Any temperature exceeding 200° F.
15.3 Oil leakage exceeding 2 cc/hour/pad.
15.4 Excessive chips or foreign material in oil.
15.5 Excessive noise or vibration during operation exceeding 95 Db, measured 10 inches from the housing.

50-Hour Load Test Procedure

1. Mount the module on a suitable load test stand in the normal mounting position. Mast should be up with the oil sump down (INPUT Pad).

2. The module requires an external lubrication system. Use MIL-L-6086, lightweight, as the lubricant.

3. Connect the three (3) inlet-oil lines to the module. Connect the two drain lines to the module.

4. Circulate room-temperature oil through the module with suitable oil pressure as determined by the no-load test procedure. Adjust the scavenge pump, so that there is no build-up of oil in the module. Circulate the oil for 30 minutes with an oil flow of 5 pounds per minute (0. 7 gallons per minute).

5. Rotate the test stand (by hand) to see if there is any binding or interferences. Then connect the interconnecting shafting and torque-up the system as per the load schedule.

6. Record the following at ten (10) minute intervals during the load operation:

   6. 1 Input speed (RPM)
   6. 2 Oil in temperature (° F)
   6. 3 Nameplate temperature (° F)
   6. 4 Oil out temperature (° F)
   6. 5 Ambient air temperature (° F)
   6. 6 Input torque (inch-pounds)

7. Photograph the test setup.

8. Load Test

   Operate the module according to the following table. Bring the module up to speed gradually for each load.

INPUT SPEED (RPM)	INPUT TORQUE (H. P.)	OPERATING TIME (HOURS)
6000	10	1
6000	50	1
6000	100	1
6000	200	1
6000	280	46

Providing the increments of time are not less than one hour, load time may be accumulated in any way suitable.

9. The module shall be rejected for any one of the following reasons:

   9.1 Any temperature exceeding 200° F.
   9.2 Oil leakage exceeding 2 cc/hour/pad.
   9.3 Excessive chips or foreign material in oil.

MATERIAL RECOMMENDATIONS

A typical preliminary design has been reviewed. Material selections were based on the information accumulated to date, plus past experience with aircraft transmissions. The selections made are not to be considered as final, but as recommendations based on our present knowledge of Cycloidal Cam Transmissions. The material list has been accumulated and tabulated in Table 11.

The employment of exotic materials in the preliminary design was discouraged. No urgent need arose for any materials other than those in common supply, and formed by good standard practices. This made for ready comparisons with contemporary transmissions.

The material and process selections reflect the following criteria:

1. Suitability for application
2. Availability of material
3. Ease of manufacturing
4. Cost.

Based on an evaluation of the working stresses, available standard materials were selected. Serious consideration was given to the fact that units would be manufactured on a prototype basis. Other materials and processes might be better recommended for quantity production of the transmissions. For example, the cam plate specification requires a carburizing grade of steel - AISI 4620 Vacuum melted. This material is obtainable and may be processed with a minimal tooling cost. If processes such as flame or induction hardening had been recommended, high tooling costs would have been incurred to make relatively few pieces. Also, any change in size of the component for transmissions of different horsepower would have resulted in new and additional tooling charges.

The carburized case depths as shown in Table 11 were selected on the basis of good commercial practice and our past experience. For most applications, experience dictates that the case depths specified on gear teeth should be approximately 1/6 of the chordal tooth thickness of the gear teeth. This has been found to be an equitable distribution between the carburized case and the core material.

The case depths specified for bearing surfaces and cam surfaces have to be deep enough to ensure that maximum shear stress below the working surface occurs in the higher strength carburized portion. The Hertz compressive stress induced by surface pressures results in an equivalent stress of opposite sign within the material. This stress should be located within the carburized area for maximum fatigue life. For example, the case depth specified on the input gear is .018 inch to .024 inch. This is the depth of carbon enrichment of the gear tooth, measured normal to the working tooth surface. With the specified case depth, the point of maximum shear in this example should occur at approximately .008 inch to .012 inch below the surface. The same general thinking applies to bearing surfaces. Ample case depth is required to assure satisfactory performance.

The cam plate, main crank bearing and bearing races have been specified to be manufactured from vacuum melted materials. This has been done to provide the highest possible fatigue life. With some processes, investigations have indicated that spalling and pitting on heavily loaded bearing surfaces resulted from inclusions in the material, some depth below the surface. The internal inclusions act as stress raisers. If these inclusions are at the point of maximum sub-surface shear, fatigue failures might ensue. Over a period of time these internal fractures work to the surface and cause a pit or spalled area to develop. The vacuum melt process eliminates material inclusions and thereby reduces fatigue failures.

A resort to exotic materials will improve the already advantageous weight/horsepower ratio of cycloid transmissions, but will increase prototype costs markedly. If it is necessary to save further weight, the materials and processes outlined in the following paragraphs will receive further consideration.

Tne use of Titanium for the cam plates necessitates flame plating of the working surfaces with Tungsten Carbide by the Linde Process.

Several titanium alloys might be considered for this application. A readily weldable alloy must be selected if fabrication is anticipated. In this event, one of the all-Alpha Titanium alloys should be used. One of the more promising weldable titanium alloys is the Ti-5Al-2.5Sn. This alloy must be welded by the heli-arc process in an inert atmosphere. Other all-Alpha alloys which can be welded are the Ti-8Al-1 Mo-1V and the Ti-6Al-4-Zr 1V.

The above titanium alloys develop tensile strengths ranging from 125,000 to 150,000 psi, and yield strengths from 120,000 to 138,000 psi. These alloys have good ductility, and an elongation ranging from 16 to 18 percent. The weld efficiencies are close to 100 percent.

For inert gas or heli-arc welding, either the tungsten or consumable electrode processes may be used. Proper surface preparation prior to welding is important. All scale, dirt and foreign material and/or other contaminants must be removed from the areas to be welded.

Other component parts of the Cycloidal Cam Transmission may be made from lightweight materials such as magnesium or beryllium. For spacers and component parts not subject to reverse bending or cyclic loading, beryllium alloys may be used. They are extremely brittle, but develop a high modulus of elasticity and are extremely light in weight.

The housing, and other parts that lend themselves to casting, could be manufactured from one of the magnesium alloys. Of the candidate magnesium alloys that might be considered, the A-Z63 and A-Z92 alloys are the most suitable.

portion. The Hertz compressive stress induced by surface pressures results in an equivalent stress of opposite sign within the material. This stress should be located within the carburized area for maximum fatigue life. For example, the case depth specified on the input gear is .018 inch to .024 inch. This is the depth of carbon enrichment of the gear tooth, measured normal to the working tooth surface. With the specified case depth, the point of maximum shear in this example should occur at approximately .008 inch to .012 inch below the surface. The same general thinking applies to bearing surfaces. Ample case depth is required to assure satisfactory performance.

The cam plate, main crank bearing and bearing races have been specified to be manufactured from vacuum melted materials. This has been done to provide the highest possible fatigue life. With some processes, investigations have indicated that spalling and pitting on heavily loaded bearing surfaces resulted from inclusions in the material, some depth below the surface. The internal inclusions act as stress raisers. If these inclusions are at the point of maximum sub-surface shear, fatigue failures might ensue. Over a period of time these internal fractures work to the surface and cause a pit or spalled area to develop. The vacuum melt process eliminates material inclusions and thereby reduces fatigue failures.

A resort to exotic materials will improve the already advantageous weight/horsepower ratio of cycloid transmissions, but will increase prototype costs markedly. If it is necessary to save further weight, the materials and processes outlined in the following paragraphs will receive further consideration.

The use of Titanium for the cam plates necessitates flame plating of the working surfaces with Tungsten Carbide by the Linde Process.

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The housing, and other parts that lend themselves to casting, could be manufactured from one of the magnesium alloys. Of the candidate magnesium alloys that might be considered, the A-Z63 and A-Z92 alloys are the most suitable.

EASE OF MANUFACTURE

General

The manufacture of Cycloidal Cam Transmissions may be accomplished by standard precision machine shop equipment and techniques. Load capacity, backlash and silence of operation are affected by, and dependent upon the accuracy of cam roller locations, the reverter pin and bearing positions, contours of the camming surfaces and the eccentric orbit of the cam plates.

All of the foregoing requirements may be met by carefully designed tooling, and the use of accurate machine tools and proper operational planning.

The cam plate contours may be produced by several different methods. Where load values are small, and backlash and noise are unimportant, the cam surfaces could be developed by form cutter milling, or by generating a Fellows-type of gear shaper. This would also be the roughing operation for more precise cams.

A precision cam requires premachining by the foregoing method, heat treating to provide the hardened wear surfaces, and subsequent grinding to the required accuracy and finish.

The accurate grinding of the cam surfaces may be accomplished in several ways, on several different types of production equipment. This would imply form grinding for a semi-precision product, or generated grinding for maximum precision. Several types of generating grinding machines are available. To name a few, by manufacturer, they are, Vinco, Gear Grinding Machine Company and Pratt and Whitney. Generating is accomplished by rotating the piece, and moving a grinding wheel in and out of the work at an established ratio.

Different generating methods are being reviewed. A form ground generating cutter in conjunction with Western Gear's Fellows' Gear Shaper Number 36Z1 is one method. Another utilizes an Illinois Tool Works high speed special Form Hob with one of Western Gear's Gould and Eberhardt hobbing machines. Other methods are also under investigation.

Anti-friction bearing inquiries have been referred to Torrington, Hyatt, SKF, MRC and RBC. Sleeve bearing materials are also under review.

Model

Experience gained in the manufacture of the cams for the display model and from subsequent discussions of other profiling techniques suggest the following cam-lobe manufacturing methods:

1. Fly-cutting with a form tool.

2. Milling, hobbing and/or grinding with a shaped wheel.

3. Generating with a form-dressed grinding wheel, where the oscillation of the wheel and rotation of the cam are timed in the correct relationship.

NOTE

The generating principle is the most meritorious where integrity of form is desired. Very satisfactory results were obtained by fly-cutting the display model cam lobes.

Hardware

A drawing of a typical cam was submitted to numerous hob and cutter tool vendors for estimated costs. The following companies have responded:

1. Barber-Colman Company
2. Allied Tool and Abrasive Supply Company
3. Illinois Tool Works
4. Union Twist Drill Company
5. Hoglund Engineering and Manufacturing Company
6. Fellows Gear Shaper Company

Other companies have responded in regard to grinding, techniques, hole locations, pins, bearings, etc. such as:

1. Equitable Engineering Company
2. Fellows Gear Shaper Company
3. Van Keuren Company
4. Kennametal Company
5. Hyatt Bearings Division
6. Roller Bearing Corporation
7. Torrington Company
8. Kaydon Company
9. Angus-Campbell, Inc.

10. Neely Enterprises
11. Ducommun
12. Edward Maltby

In the 3 cam-plate, single-stage design of the type depicted in Drawing 6144R24, it will be necessary to maintain extreme accuracy of form and spacing of the reverter holes, pins and fixed pins. The techniques adopted for this preliminary design assist in maintaining such accuracy by:

1. Fabricating the individual cam rotor assemblies from standard cam plates and alloy hubs, enabling the 3 cam plates to be form finished as a stacked assembly. After finishing, the individual cam plates are to be secured to the hubs.

2. The spacing of the fixed pins, also the reverting pins, would be achieved by the gage pin and block construction shown in the section in the center of the referenced drawing. The gage pins and blocks, with an order of accuracy of 10 millionths inch, are locked together by "shrinking on" the outer ring. By this method, jig boring of holes for pin location would be avoided, except for the boring of the reverting holes in the cam plates. Other methods for producing these holes are under consideration, as are other means of fixed and reverting pin location.

NOTE

The gage pin and block method discussed under Item 2 was later discarded. A preliminary weight analysis proved the design was too heavy.

MAINTENANCE

An analysis of a typical Cycloidal Cam Transmission design was made in an endeavor to disclose maintenance problems that may be created by malfunctions or failures and periodic examinations and overhauls.

The basic concept of a Cycloidal Cam Transmission consists of a multiplicity of similar elements suggesting a certain amount of interchangeability. The moving elements subject to wear or failure are the cam plates, anti-friction bearings, shafts and seals. Because operating clearances or interferences are established in manufacture there is no provision or requirement for field adjustment. By eliminating the necessity or probability of field adjustment, maintenance is simplified.

Field or Base maintenance need only entail the operations to dismantle, clean, inspect, replace components (if required), reassemble, lubricate and test.

Maintenance Tooling may be very elementary. It might include holding fixtures, standard wrenches and tools, torque wrenches, a forcing press for bearings and seals, a run-in test stand and the usual cleaning and parts storage facilities.

STUDIES OF TURBINE POWER TAKE-OFFS

In the past, difficulties have been experienced in the application of conventional gear reduction systems to gas turbine engines due to the effect of high frequency vibration on gear life. The Cycloid Cam Transmission may overcome the deficiencies found in some gear transmissions. Inherently, the drive elements make only rolling contacts and sustain them with no interruptions. This filter action of the cycloid transmission, notable for smooth power conversion, aids its adaptation for gas turbine engine application.

The six gas turbine engines considered here offer a cross section of the speed and horsepower ranges of interest.

Table 6 and 7 (Power Source Analysis) give applicable data, and Figures 27 to 30 inclusive, show the basic installation dimensions and power take-off points of these engines.

PARAMETRIC CHARTS AND GRAPHS

Four graphs are included in this section, namely:

Figure 34 Weight vs Horsepower with Reduction Radio as a Parameter

Figure 34 provides a comparison with known conventional gearing systems. The curves show a definite advantage for the Cycloidal Cam Transmission in the area of weight vs reduction ratio.

Figure 35 Weight vs Reduction Ratio with Horsepower as a Parameter

Figure 35 discloses that in the modular design concept, the weight penalty for increased ratio is insignificant in the lower horsepower ranges, and amounts to only a 5 percent increase in weight as the reduction ratio increases from 19:1 to 150:1, in the 2520 horsepower range.

Figure 36 Weight/Horsepower vs Input Speed with Horsepower as a Parameter

Figure 36 shows that over a wide range the change in the "Pounds per Horsepower" ratio for the selected transmissions is insignificant.

Figure 37 Weight vs Input Torque with Reduction Ratio as a Parameter

Figure 37 is a useful guide for establishing approximate unit weights when the input speed and power are known.

COST ESTIMATES

The preliminary designs included in this report are purely basic and reflect the design principle more than the economical approach to the problem.

Nevertheless, these designs have been cost estimated, and the estimates have been included in the report for reference. It is our opinion that these estimates will change as more design information becomes available in Phase II of the program.

A cost breakdown for the 280, 840 and 2520 horsepower unit is tabulated below. Reference is also made to Figures 38, 39, 40 and 41, which are graphical plots of cost vs horsepower, tooling and test costs vs horsepower, cost per pound vs weight and cost per pound vs horsepower, respectively.

Horsepower	Unit Cost (Dollars)	Tooling Cost (Dollars)	Testing Cost (Dollars)	Total Cost (Dollars)
280	18,850*	11,466	3,740	34,056
840	63,690*	15,613	4,668	83,971**
2520	189,911*	27,800	19,710	237,421

* The unit cost would be reduced for quantity production.

** Additional cost savings, possibly 10 to 15 percent, would be realized by first fabricating the 280 horsepower unit and then fabricating an 840 horsepower unit.

NOTE

The 280 horsepower estimate is based on Drawing 6144R43, and is the building block for the higher horsepower range. Therefore, by careful and simplified designing, and relaxation of tolerances that are not critical to the functioning of the transmission, considerable cost savings can be realized. A more comprehensive cost estimate will be made during the later half of Phase II.

EVALUATIONS

Review of Gear-Type Transmissions

Some examples of Western Gear Corporation's aircraft background in the design, manufacture and testing of lightweight, highly loaded gearing and gear applications are shown in the following list:

HILLER AIRCRAFT COMPANY

WGC NO.	Hiller Model	Helicopter Applications
1477	H23-B H23-C	Main Rotor Transmission
1486	H23-B H23-C	Tail Rotor Drive
1067	H23-B H23-C	Tail Rotor Drive
1962	H23-D	280 Horsepower Main Transmission
1014	H23-D	Tail Rotor Drive

HUGHES TOOL COMPANY

WGC NO.	Hughes Model	Helicopter Applications
1587	H-17	Accessory Drive for the Electrical, Hydraulic and Lubrication Systems
1181	HO-2HU	Main Transmission

NORTHROP

WGC NO.	Northrop Model	Aircraft Applications
1470	Turbodyne	10,000 Horsepower Turbo-Prop Drive and Accessory Transmission

SOLAR AIRCRAFT

WGC NO.	Solar Model	Aircraft Applications
1506	Mars	Accessory and Pump Drive for their Turbo-Input Speed 40,000 rpm
1620	Mars	Airborne APU System with 40,000 rpm input speed
1922	Jupiter	Reduction and Accessory Drive, incorporating eight (8) pads
1431	Jupiter	Auxiliary Power Reduction and Accessory Drive
1431	Jupiter	Combining Box and Accessory Drive

The H23-D transmission is the first helicopter transmission ever designed and developed by a gear manufacturer. Formerly, designs of this nature had been done by the helicopter manufacturers themselves.

Some of Western Gear Corporation's early helicopter transmission experience was gained through the production and manufacture of transmissions and tail rotor drives for Hiller, Model H23-B.

In 1949, Western Gear Corporation engaged in the manufacture of the main transmission and tail rotor drive gearbox for the Hiller H23-C, three-place helicopter. In the latter stages of our participation in this program, we re-designed the mounting of the input sun gear in the main transmission, which increased the overhaul life of the transmission approximately 25 percent.

In 1955, we designed the main transmission and tail rotor drive gearbox for the Hiller H23-D helicopter. This transmission was designed for an input horsepower of 280 and was approximately 25 pounds lighter than the H23-C model transmission which was rated at 150 horsepower input. The Model H23-D transmission was first designed for a 1,000 hour overhaul life. The dry weight was 122 pounds with a reduction ratio of 9:1. Two-stage planetary gearing was used with three accessory pads. We participated in the extensive qualification test program conducted by Hiller on the drive system, and also manufactured the transmission and tail rotor drive gearbox. The Model H23-D transmission represented the first major step toward obtaining helicopter utility and performance not financially overshadowed by unreasonably high maintenance burdens and operating cost.

Western Gear Corporation manufactured the main transmission and tail rotor drive gearbox for the Hughes YHD-2HU, two-place helicopter, and also the complete drive system, (including main gear box, torque tubes and wing gear boxes) for the Doak Model 16 VTOL Research Airplane. Other experience includes manufacture of transmission components for McDonnell, Ryan and Lockheed.

Some typical transmissions and test set-ups are shown in the Photograph section of the report.

Patent Search and Evaluations

A visit was made to the U.S. Patent Office in Washington, D.C. on July 21, 1960. A selected group of patents relating to cam arrangements were chosen for evaluation. Since that time additional patents have been reviewed. The patents have been tabulated and follow.

Several different patent variations of the basic cycloidal cam principle have been reviewed, with the intent to define a qualitative Figure of Merit for a basis of comparison. The ratings and scoring methods are displayed in Table 12. These evaluations are not intended to be conclusive but are submitted as qualitative reflections on recorded designs of patents on file in the U.S. Patent Office.

NOTE

The highest attainable rating is 35. The pure cycloidal cam (Patent 1,694,031) placed third with a rating of 28.

List of Patents

Patent Number	Title, Inventor and Date
17,811	Gear Transmission - L. K. Braren - September 23, 1930
24,288	Speed Changing Device - A. V. Nanni - March 19, 1957
458,677	Transmission Gears - R. Candone - July 25, 1950
498,552	Rotary Cam Gear - C. W. & W. F. Hunt - May 30, 1893
500,322	Differential Motion Mechanism - J. Dronsfield - June 27, 1893
500,332	Drive Chain - C. W. Hunt - June 27, 1893
672,263	Portable Pneumatic Drill - W. E. Dean - April 16, 1901
834,799	Transmission Gears - W. D. Bensinger, A. Wente - February 21, 1952
1,116,970	Power Transmitting Mechanism - V. G. Apple - November 10, 1914
1,449,352	Gearless Planetary Transmission - F. W. Seeck - March 20, 1923
1,543,791	Transmission Gearing - W. C. Pitter - June 30, 1925
1,641,766	Speed Reducer - A. Laukhuff - September 6, 1927
1,692,160	Mechanical Movement - J. A. Dormer - November 20, 1928
1,694,031	Gear Transmission - L. K. Braren - December 4, 1928
1,738,662	Ball Transmission - G. S. Morison - December 10, 1929
1,767,866	Gearing - E. Wildhaber - June 24, 1930
1,773,568	Gear Transmission - L. K. Braren - August 19, 1930
1,828,795	Form of Reduction Gear - G. W. C. Webb - October 27, 1931
1,867,492	Gear Transmission - L. K. Braren - July 12, 1932
1,870,875	Speed Reducing Transmission Device - P. Scheuer - August 9, 1932
1,910,777	Epicyclic Gearing - F. Soddy - May 23, 1933
1,942,795	Power Transmission and Speed Reduction System - M. B. Benson - January 9, 1934

Patent Number	Title, Inventor and Date
2,239,839	Differential Gearing - M. B. Benson - April 29, 1941
2,475,504	Reduction Gear - J. A. Jackson - July 5, 1949
2,508,121	Gear Transmission - W. K. McIver - May 16, 1950
2,520,282	Speed Reducing Power Transmission - E.W. Henry - August 29, 1950
2,529,997	Epicyclic Drive - L. H. Browne - November 14, 1950
2,666,345	Speed Reducer - W. E. Amberg - January 19, 1954
2,677,288	Speed Reducing Transmission Mechanism - A. Gnahrich - May 4, 1954
2,874,594	Speed Reducer - E. V. Sundt - February 24, 1959
2,929,265	Strain Wave Gearing - Multiple Tooth Differences - C. W. Musser - March 22, 1960
2,929,266	Strain-Wave Gearing-Tubular Shaft - C. W. Musser - March 22, 1960
2,930,253	Wave Generator - C. W. Musser - March 29, 1960
2,930,254	Harmonic Gearing with Extended Contact - C. W. Musser et al - March 29, 1960
2,931,248	Strain Wave Gearing - Strain Inducer Species - C. W. Musser - April 5, 1960
2,931,249	Strain Wave Gearing - Bearing Variable Elements - C. W. Musser - April 5, 1960
2,932,986	Strain Wave Gear-Species in Which Only One of the Gears is Input - C. W. Musser - April 19, 1960
2,943,513	Dual Strain Wave Gearing - C. W. Musser - July 5, 1960

Western Gear Corporation's Patent Application Descriptions

Three ideas have been developed which reasonably appear to be patentable. These ideas are:

Docket 4168 Roller Transmission
Docket 4169 Right Angle Cam Ring Transmission
Docket 4170 Cycloidal Cam Transmission

On the recommendation of our Patent Attorneys, Docket 4168 has been combined with Docket 4170. The net result is a revised patent description. The revised version is included in this section of the report. In addition, the patent description for the Right Angle Cam Transmission has also been included.

Revised Docket 4170 - Cycloidal Cam Transmission

1. This invention relates to a means of producing a changed speed rotary motion on a common axis without the use of gears or other means commonly used to accomplish the purpose.

2. In this invention, the required changed speed rotary motion is accomplished without gearing and without the use of sliding surfaces. The speed change is produced by a novel arrangement of camming surfaces operating on reaction rollers with the generated interference between cams and rollers resulting in rotative motion.

3. The reaction and consequent changed speed rotary motion is created by causing the reacting cam discs to describe a cycloidal path within a fixed cage containing the reaction rollers.

4. Through the employment of a novel arrangement of mechanical elements, the gyratory rotary motion of the cam discs is converted to a smooth, continuous rotary output motion.

5. Because all action is performed without use of sliding surfaces, frictional losses are reduced to a very low value, permitting relatively great powers to be transmitted with comparatively small elements and with a minimum of lost motion or backlash.

6. The construction permits of a large ratio change from input to output speeds in a single stage with balanced dynamic forces. This is of great importance when high input speeds are involved. This is accomplished without the use of small, highly loaded members such as would obtain in a conventional gear transmission or gear train.

7. An object of the invention is to provide a reduced speed rotary motion from a high speed prime mover to the rotary wing system of a helicopter air vehicle without the use of gearing.

8. Another object of the invention is to provide a reduced speed rotary motion from a high speed prime mover to the propeller of a boat or ship without the use of gearing.

9. The invention is not limited to the uses recited, but can be used to supply a changed speed rotary motion to any device requiring such motion.

10. The invention is not limited to use as a speed reducer, but can be used as a speed changer with various ratios.

11. The ratios available from a selected combination of numbers of cam lobes and reaction surfaces or rollers is dependent on the element selected for power input and the element selected to be fixed or held stationary.

12. Other objects and advantages of this invention will be apparent from the following detailed description of a preferred embodiment thereof as illustrated in the accompanying drawings.

13. Figure 1 is an isometric drawing of a cycloidal cam transmission with sections cut away to show relationship of the internal parts.

14. Figure 2 is an isometric drawing of a cycloidal cam transmission as shown in Figure 1, but with parts "exploded" for clarity of presentation.

15. While the mechanism is detailed, it is to be understood that this invention is not limited in its applicability to a type mechanism as illustrated. The 4 basic elements (eccentric, cam discs, reaction cage, reverting system) can be combined

in various combinations to provide different resultant ratios depending upon the elements selected for input and output, and the element selected to be held stationary to provide the reactive component.

16. Referring to the Drawing Figure 1 by characters of reference, it will be seen that: the object of this construction is to cause output members A_1 and A_2 to be rotated at a speed reduced from that of input eccentrics B_1, B_2, B_3. In the configuration this is accomplished as follows:

17. Input eccentrics B_1, B_2, B_3 are attached to a shaft (C). Rotation of this shaft with its attached eccentrics B_1, B_2, B_3 causes movement of cam discs D_1, D_2, D_3, which are mounted on the eccentrics to permit rotative motion. The rotation of the eccentrics cause cam discs D_1, D_2, D_3 to describe a gyratory orbit around the axis of the input shaft. Engageably positioned around the lobes of the cam discs are ten (10) reaction shafts (rollers) (E). These shafts/rollers are contained in a reaction cage F_1, F_2, F_3 and are radially disposed around the axis of the input shaft. The cycloidal movement of the 9 cam disc lobes causes a wedging action to occur between cam lobes and the reaction rollers.

18. The wedging action of the cam lobes causes a rotary reactive component motion to be exerted on the cam discs. This rotary motion is cycloidal, but continuous in the same direction and is equal to the amount of one lobe pitch for one revolution of an eccentric, but opposite in direction to that of the eccentric. In the design shown there are 9 cam lobes reacting on 10 reaction rollers. In consequence, 9 revolutions of the eccentrics are required to produce one revolution of cage F_1, F_2, F_3 or a ratio of 9 to 1.

19. The rotation of cam discs D_1, D_2, D_3 while continuous, is of a cycloidal nature. This gyratory rotation is reverted to continuous, smooth motion in the following manner: - Cam discs D_1, D_2, D_3 are provided with reverter bores (G). Reverter shafts (H) extend through the reverter bores (G). Reverter shafts (H) have eccentrics (I) attached. Eccentrics (I) have a throw equal to the throw of eccentrics (B) on the input shaft (C).

20. Because of the cycloidal action of the cam discs, the reverter bores describe a gyratory path around the axis of

the reverter shafts (H). This cycloidal motion of the reverter bores causes the reverting eccentrics (I) to rotate at the same speed and in the same phase as input eccentrics (B). This cycloidal action of reverter bores on reverting eccentrics (I) reverts or changes the pulsed rotation of the cam rings to a smooth, continuous rotary motion of output members A_1, A_2. The reduced speed rotary motion of A_1, A_2 is the output speed to supply the required reduced speed to a device.

21. When the input speeds to the cycloidal speed changes are high, it is desirable to balance the dynamic forces generated due to the eccentric rotation of the cam discs. In the design shown such dynamic forces are balanced in the following described manner:

22. Three cam discs are employed: D_1, D_2, D_3. D_1 and D_3 are located on each side of D_2 and are 180° out of phase with D_2. The mass of D_2 is equal to the combined mass of D_1 and D_3. Dynamic forces of the discs are balanced because when the mass of D_1 and D_3 is moving away from the axis of the system a like mass (D_2) 180° opposite is likewise moving outward, thus the force is balanced. Conversely, masses are balanced when moving in towards the system axis. Thus, dynamic unbalance of the discs is neutralized.

23. This principle of dynamic balancing is not limited to the use of three discs. Groups of three discs each may be employed when required to transmit greater horsepowers. Also groups of two discs each may also be employed for lower input speeds where auto balance is not required.

24. In operation, the entire input system comprising input eccentrics B_1, B_2, B_3, Shaft (C), cam discs D_1, D_2, D_3 and reverter shafts (I) form a phase locked system.

25. The phase relationship is caused and maintained by the similarity of eccentric throw of input eccentrics (B) and the kinematics of the action of cam plates (D) acting on reaction rollers (E).

26. This phase relationship is further assured by the action of cam plates (D) on the reaction rollers (E).

27. Rotation of input eccentrics (B) causes a like rotation of reverter shaft (I), thus cam plates (D) are maintained in phase relationship.

28. The gyratory motion of the cam plates (D) cause bores (G) to describe an eccentric orbit around the axis of shafts (H).

29. The rotation of the cam plates (D) in their gyrations around the input shaft (C) cause reverter shafts (H) to rotate on their axes at a uniform radius around the input shaft (C) axis, but at a reduced, constant speed.

30. The reverter shafts (H) rotate in bearings (K) mounted in output plate A_2.

31. Bearings (K) are mounted in cam plate bores (G) and bear on eccentrics (I).

32. The resultant of forces acting on eccentrics (I) cause a rotation of shafts (H) in output plate bearings (K).

33. The forces acting on eccentrics (I) load the shafts (H) between cam plates D_1 and D_2 and between cam plate D_3 and output plate A_2 in double shear.

34. Flange A_1 is part of, or attached to, output plate A_2, thus providing means to attach a load requiring increased torque and reduced speed from that of input shaft (C).

35. It will be noted that reverter shafts (I) drive output shaft A from one end only.

36. The phase locked feature of the system removes bending forces from shaft (H), thus permitting the use of a drive element at only one end of shaft (H).

37. The single end drive is only possible because eccentrics (I) are employed to fill the bores (G) in cam plate bearings (K) mounted in cam plates D_1, D_2, D_3 in bore (G).

38. By the employment of an eccentric in each reverter bore a further improvement is achieved.

39. This consists of permitting and causing each reverter pin to share a proportionate amount of the load torque developed

through the cam/reaction pin system.

40. Such load sharing permits much greater torques to be transmitted than would obtain if reverter eccentrics were omitted.

41. Should eccentrics be omitted, the reverter pins would only contact the reverter bores during one-half of the eccentric path that each reverter bore describes around the reverter pin.

42. During the remaining half of the eccentric path the reverter pin would be free from reverter bore contact, and thus the torque load capacity of the reverter system would be greatly reduced.

43. In such a system the loading on the reverter pins during the half time contact phase rises from zero to a maximum and then relaxes to zero, periodically.

44. In effect, such an arrangement would cause one-half of the reverter pin system to be subjected to the full torque loading in varying amounts.

45. This would require employment of very much larger reverter pins and supporting structure for a given torque load.

46. The use of an eccentric in each reverter bore permits and causes each reverter pin to share the load equally, thus achieving a more compact and reliable design for a given load.

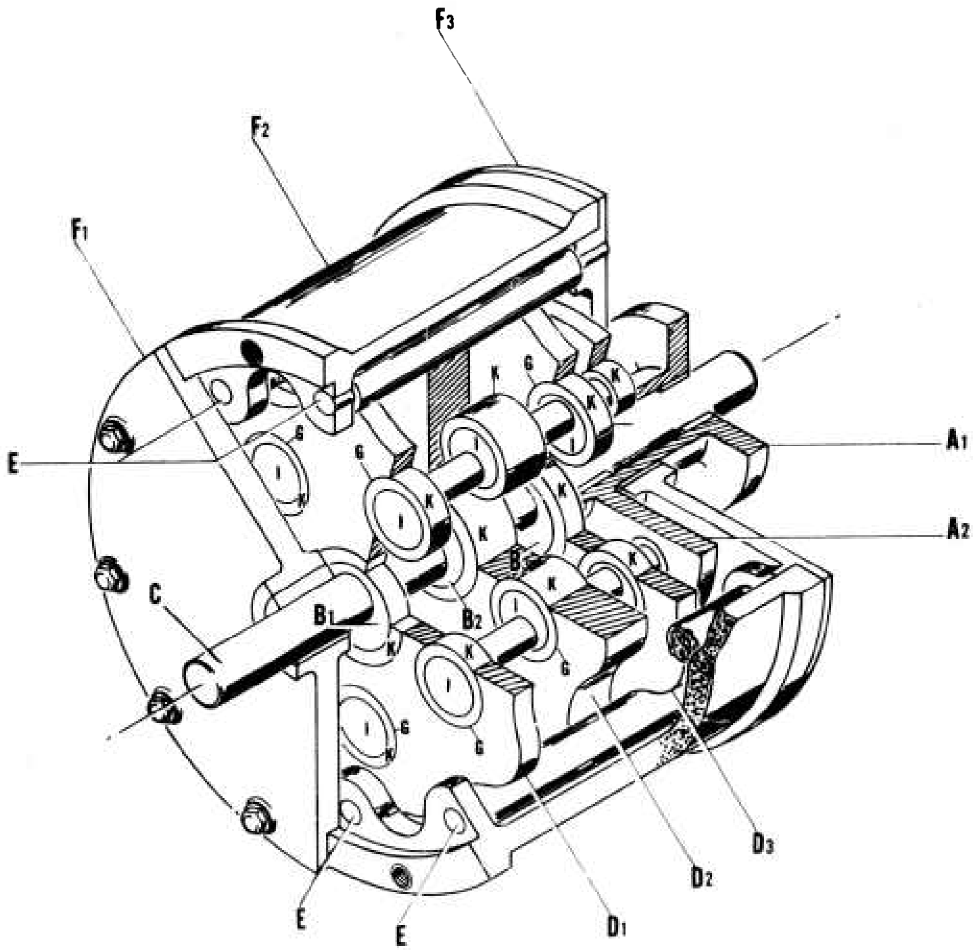
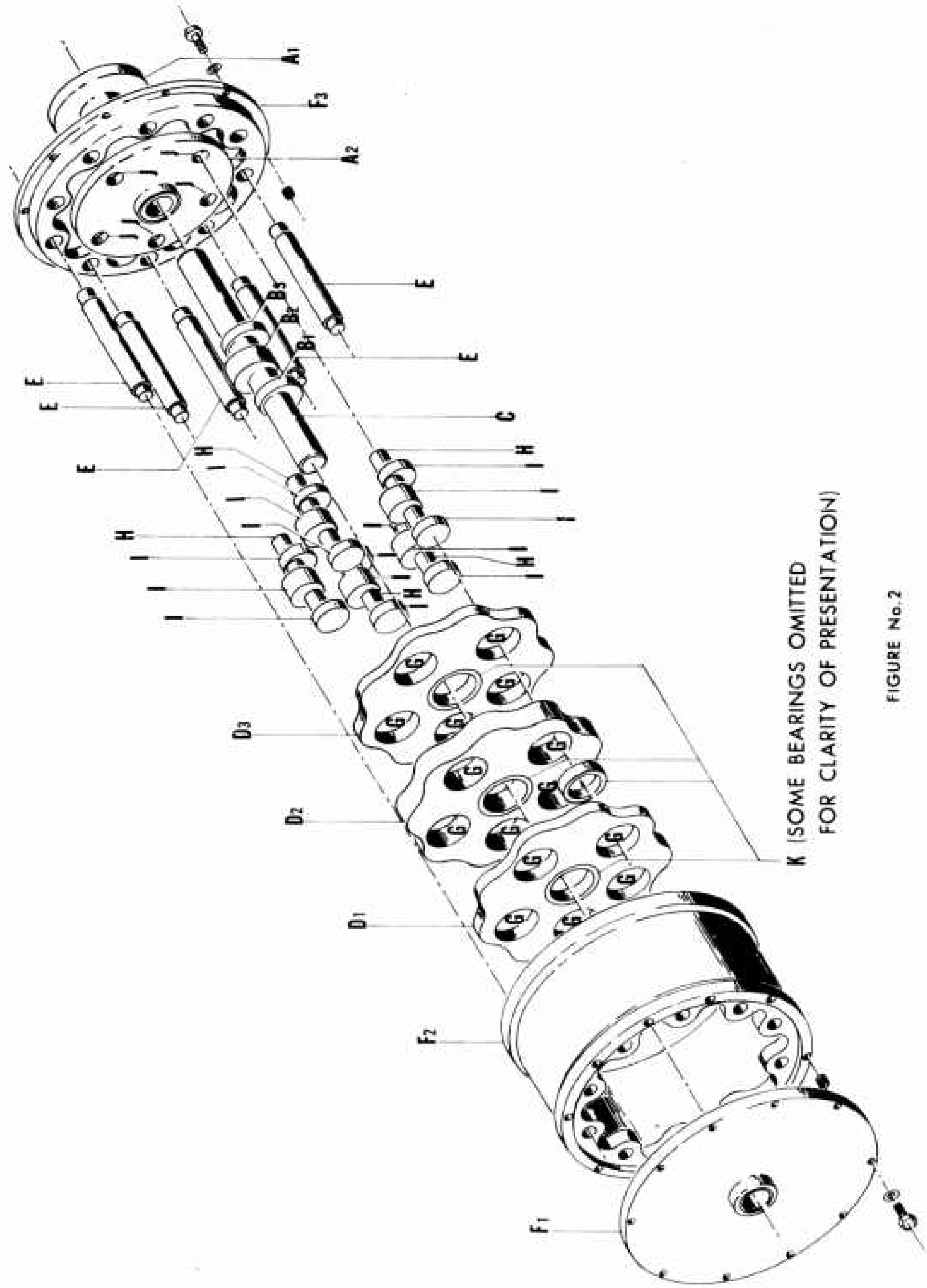


FIGURE No. 1



A1
F3
A2
E
B3
B2
B1
C
H
I
G
D3
D2
D1
F2
F1
K (SOME BEARINGS OMITTED FOR CLARITY OF PRESENTATION)

FIGURE No.2

1. This invention relates to a means of producing a rotary motion to a shaft from a rotary motion substantially at right angles from the output shaft axis without the use of gears or worm gearing or other means commonly used to accomplish the purpose.

2. In this invention the required right angled rotary motion change is accomplished without gearing and without the use of sliding surfaces. The motion transition is produced by a novel arrangement of inclined planes operating on rollers and in consequence the backlash or lost motion of the system is kept at a very low value and frictional losses are very small due to the complete absence of sliding surfaces.

3. In the invention all great loads on the system are carried by compression members and shear and bending stresses are almost completely eliminated.

4. The invention permits relatively great ratio change from input speed to output speed without the use of small, highly loaded members such as would obtain in a gear-to-gear or worm-to-gear right-angled motion transducer.

5. An object of the invention is to transduce rotary motion from a prime mover with a horizontally disposed shaft axis to a vertical shaft supplying rotative action to a rotary wing of a helicopter air vehicle with better efficiencies than can be attained by use of gearing commonly used for the purpose.

6. Another object of the invention is to provide right-angled reduced motion to any device requiring a right-angled transduced motion with low frictional losses or low back-lash or rugged construction or a combination of these features.

7. The invention is not limited to the uses recited but can be used to supply right-angled rotary motion to any device requiring such motion.

8. Other objects and advantages of this invention will be

apparent from the following detailed description of a preferred embodiment thereof, as illustrated in the accompanying drawings.

9. In the drawing: Figure 1 is an isometric drawing of a helicopter right angle speed reducing transmission.

Figure 2 is an isometric drawing of the contained motion transducing mechanism by which the right angle motion change is accomplished.

10. While the mechanism is detailed, it is to be understood that this invention is not limited in its applicability to a type of mechanism as illustrated.

11. Referring to the drawings 1 and 2 by characters of reference, it will be seen that: the object of this construction is to cause the power output shaft (A) to be rotated at a reduced speed by power input shaft (B). In the configuration shown, this is accomplished as follows:

12. Rotation of input shaft (B) rotates attached eccentrics (C) and (D). It will be seen that eccentric (C) is 180° out of phase with eccentric (D). A cam ring (E) is located on eccentrics (C) and (D) through spherical self-aligning bearings (F). Rotation of shaft (B) rotates the attached eccentrics (C) and (D) within the spherical bearings (F) and causes the cam ring (E) to oscillate around the axis of shaft (A). In the configuration shown cam ring (E) has seven (7) lobes. Engagably positioned above cam ring (E) is a reaction roller cage (G) carrying eight (8) reaction rollers (H). This reaction roller cage is attached to output shaft (A).

13. The before described tilting and oscillatory motion of cam ring (E) creates an interference between cam ring lobes and reaction cage rollers (H). Because the cam lobes are in effect inclined planes such interference causes a rotative component to be applied to the reaction roller cage and attached shaft (A). This is in part due to the tilting action imparted to the cam ring (E) and in part to the oscillatory motion of the cam ring. The resultant rotary motion of the output roller cage creates a further interference between cam ring lobes and reaction rollers. This interference causes the cam ring to tilt around the

axis of shaft (B). This tilting is caused by the rotative motion of the reaction rollers reacting on the cam ring lobes and the tilt causes the cam lobes to permit a greater rotative movement of the reaction roller cage to occur.

14. During the tilt/rocking action of cam ring, the cam ring is partially rotating around shaft (A) axis. This causes more partial rotation of the reaction roller cage (G). It will be thus seen that continuous rotation of shaft (B) causes the cam ring (E) to tilt or rock along the axis of shaft (B) and at the same time rock around the axis of shaft (B) and to oscillate around the axis of shaft (A). The result of these combined motions causes a continuous rotary motion to be imparted to shaft (A) but at a speed reduced from that of shaft (B).

15. In the configuration shown, seven (7) revolutions of shaft (B) are required to produce one (1) revolution of shaft (A) or at a ratio of 7:1.

16. The ratio change of this invention is not limited to the example cited but can be any ratio required, provided that there are one more in number of reaction rollers than number of cam ring lobes.

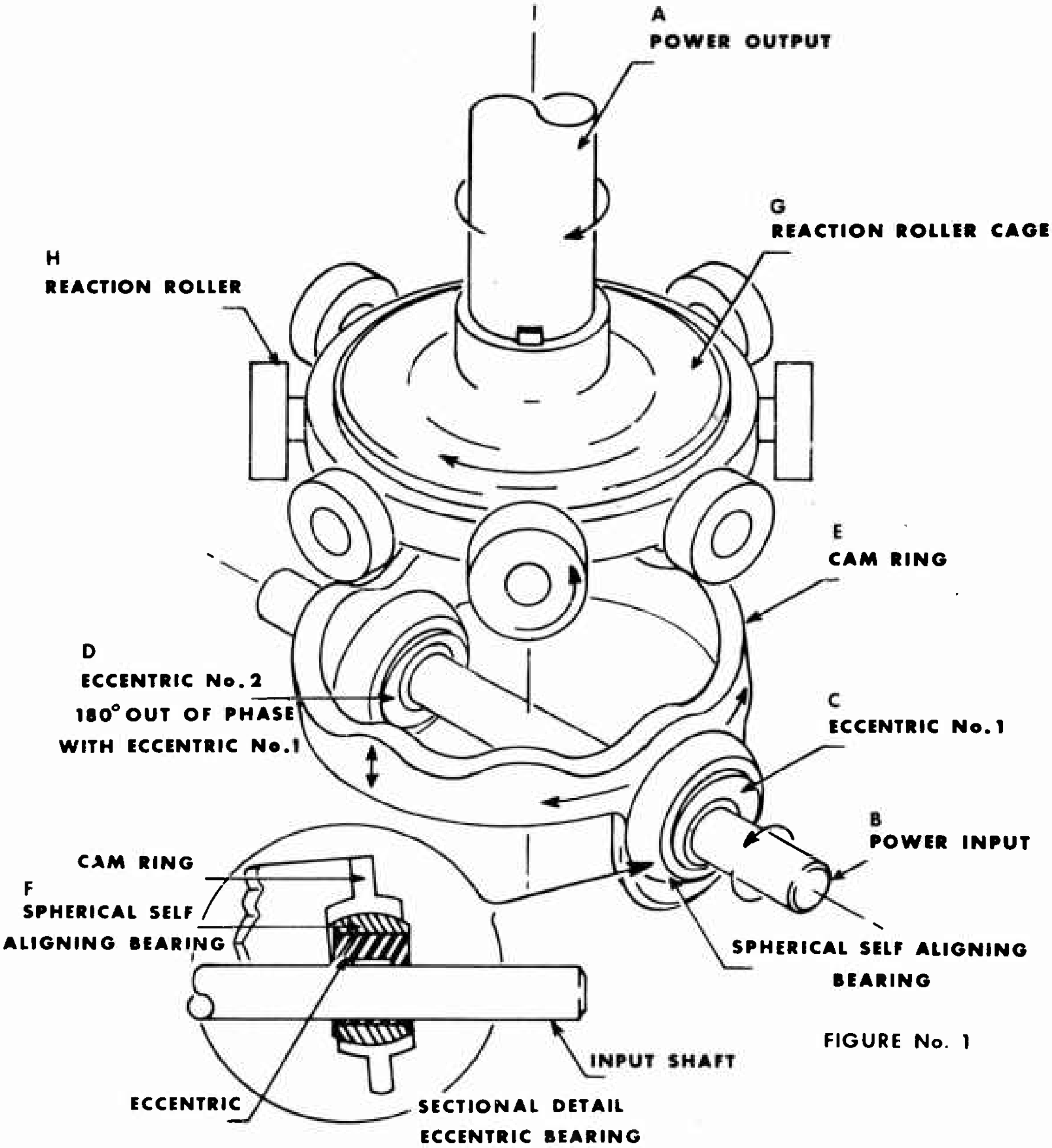


FIGURE No. 1

FIGURE No. 2

Helicopter Design Guide

Western Gear Corporation's experience in Army helicopter transmission engineering has been made available, exposed to review and duly classified and noted, where applicable to this study.

The prime Military Specifications governing the general requirements for helicopter transmission systems and helicopter ground testing have been reviewed. These specifications are as follows:

MIL-T-5955A (Amendment 1)
MIL-T-8679

The establishment of other design criteria was referred to Western Gear Corporation's practice in the design of the Army's Raven (H-23D) helicopter 280 horsepower transmission. The accompanying data sheet summarizes the maximum requirements in critical sections of the main rotor drive.

Data Sheet

Bearing Life
B-10 Life of 1000 Hours

Load Schedule
Full Rated Power - 15%
90% Rated Power - 35%
75% Rated Power - 50%
± 25% Superimposed Torsional Oscillatory Load

Strength Criterion
Peak Load
2X (Full Rated Power)

Weight/horsepower - 1.0/2.2 = .455 pound/horsepower

Torque/weight - 374/1.0 inch-pounds/pound

Dry weight - 122 pounds

Flight Loads
Average
Vertical - 4020 pounds
Horizontal - 425 pounds

Main Bearing Loads - Hertz Loading
Lower Mast - 142, 500 psi
Upper Mast - 248, 000 psi

Gear Tooth Loads
First Stage Planetary

	Life Cycles	Root Stress	Compressive Stress	PVT
Pinion	6.5×10^6	39, 500	140, 000	291, 000
Gear	10×10^6	14, 000	208, 000	550, 000

Maximum Mast Loads
Torsional - 28, 200 psi (max)
Tensile - 72, 000 psi (max)
Combined - 86, 900 psi (max)

Overall Gear Ratio
9:1 Two Stage Planetary System

Input Speed
2900 rpm

Rotor Speed
322 rpm

Bearing Survey

Singular to cyclodial cam transmissions, due to the pure rolling action of the drive elements, is the importance of the allowable Hertz Stress. A survey was made of anti-friction bearing practices in the ball and roller bearing industry.

For this purpose, bearings of the Deep Groove Ball, and non-separable roller types were analyzed on their catalogue ratings for allowable Hertz stresses. Tables number 13 and number 16 reflect these and the bearing identities. The data reduction to Hertz stress for each series of the different types was averaged

An S-N plot was drawn for a bearing in each series which most nearly approximated the average Hertz stress for the series. This data is tabulated in Table 14 and 15 and graphed in Figure 33.

These results are conclusive in that they apparently reflect the bearing industry's safe practice and are in agreement with the findings of T. Barish of Marlin-Rockwell Corporation (MRC), a recognized authority. His paper reports on the rating of 206 ball and roller bearings made by different manufacturers compared at a rating of 1370 pounds at 500 rpm, and 500 hours B-10 life. The recorded Hertz stresses at these ratings are:

Ball Bearings	443,000 psi
Roller Bearings	324,000 psi

The S-N plot of Figure 33 reflects these values very closely.

Investigation of materials in the high strength domain is being conducted concurrently by Western Gear Corporation's Research Department. The dash line of Figure 33 represents a leading bearing manufacturer's success in improving the surface endurance limits of a bearing steel by a special heat treating process.

Further exploration is being made of processes for materials which will extend the design limit.

PLANS FOR CONTINUING PERFORMANCE

During the course of our investigation into the Cycloidal Cam Transmission and its possibilities, it was inevitable that many new ideas would be generated. Some represent radical departures from the original concept.

Three (3) of the more promising ideas were reviewed. Brief engineering studies have been made and are incorporated in this report. These ideas are as follows:

1. The substitution of rollers in place of cams to produce the required cycloid action. The idea is illustrated in Figure 42.

2. A single stage compounded cycloid cam employing the differential principle to achieve a two stage reduction. Drawing 6144E42 depicts this system.

3. A right angle cam transmission for horizontally disposed engines with vertical rotors. Figure 43 shows the basic configurations.

The results of the studies indicate that items 1 and 2 above have merit for some applications, but not for the 280 horsepower and 840 horsepower transmissions as required for Phase II of the contract. In consequence it is recommended that no further studies be made on these ideas. Item 3 above offers considerable promise for Army aircraft applications. Therefore, it is recommended that a more comprehensive study be conducted. It is also recommended that a model be produced so that the kinematics can be visualized and demonstrated.

Right Angle Cam Transmission

The design (Figure 43) consists of a shaft (power input) having two eccentrics attached and 180 degrees out of phase. These eccentrics have self-aligning spherical type bearings riding on their peripheries. These bearings are positioned in the bores of a cam ring at right angles to the cam surfaces. The cam is a ring section with cam surfaces on the face of the ring. Such cams are generally radial to the axis of the ring. Rotation of the eccentric causes a rocking back and forth along the eccentric shaft axis. The cam ring is free to rock back and forth around the shaft axis.

The cam surfaces engage with rollers on a reaction roller cage. This member consists of a disc-type cage mounted on an output shaft. The cage has rollers radially located around the edge of the cage and with roller axis at right angles to the output shaft axis. This is also the axis of the cam ring.

The reaction roller cage has one more roller than the number of lobes on the cam ring. Action is as follows:

Rotation of the eccentric shaft causes the cam ring to rock and tilt up and down along the eccentric shaft axis. The cam ring is free to rock or tilt around the eccentric shaft axis.

Such rotation of the eccentric shaft, and cam ring tilting, creates interference between cam surfaces and reaction rollers. This interference causes a partial rotation of the reaction cage around the output shaft axis. At the same time the reaction roller movement causes interference between the other cam surfaces and reaction rollers. This forces the cam ring to tilt around the eccentric shaft axis. The tilt reduces interference between surfaces by causing further partial rotation of the rollers.

In effect, the cam ring wobbles on the output shaft axis and because of this action and also because of the difference in number between cam lobes and reaction roller, rotary motion is imparted to the reaction roller cage. The rotation is at a reduced rate from the eccentric shaft rotation. The rate of ratio is dependent on the designed number of cams and rollers, ie, if there are 18 rollers and 17 lobes (a difference of 1) the ratio would be $\frac{18}{18-17}$ or $\frac{18}{1}$ or 18:1.

Model of The Cycloidal Cam Transmission

The display model has ably demonstrated the validity of various theories concerning the cycloidal cam type of transmission.

Apart from the smoothness and quietness of action with no appreciable backlash, the model has demonstrated the ability of cycloidal systems to parallel the Epicyclic Gear Systems in as much that the various members, locked or free to turn, in certain combinations, give different output ratios. These combinations and ratios are described in the "Cycloidal Cam Configurations" section of this report.

Further experimentation with the model is recommended as listed below:

1. Replacc the lucite cam plates with metal cam plates and conduct simple loading tests.

2. Attach "strainline photoelastic" gages to the cam plates in order to observe the loading kinematics during testing.

3. Installation of anti-friction bearings and/or sleeve bearings to allow momentary input speeds of 6000 RPM.

4. Take slow-motion movies of the model during the experimental testing.

280 Horsepower Transmission

It is planned that a finalized design of the basic module be processed during Phase II. The preferred ratio would be 19:1 with an input speed of 6000 RPM. The necessary shop drawings will be released for the manufacture of two (2) experimental modules during Phase III. Also, it is recommended that no load experimental testing be conducted followed by 50 hour torque load testing. For comparison with the noise levels of present geared transmissions, sound level studies should be made during the test program.

840 Horsepower Transmission

Based on the modular concept, an 840-horsepower transmission consisting of three (3) modules would be designed, but not detailed, for manufacture. The three modules would be arranged in a clover-leaf pattern. The design would be with an input speed of 6000 RPM and an overall reduction ratio of 29:1. The input combining gear box would have a ratio of 1:1 with the output combining gear box having a ratio of 1.52:1.

Reliability Testing

After the two (2) 280-horsepower modules have been "debugged" and the necessary changes made, it is recommended that a minimum of eight (8) modules be released for manufacture. These modules would be shipped to U.S. Army (TRECOM) for reliability testing, using their test stand. The testing would be witnessed by Western Gear Corporation representatives.

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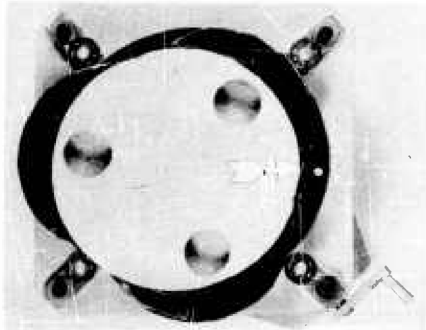
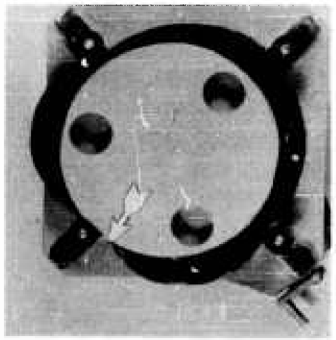


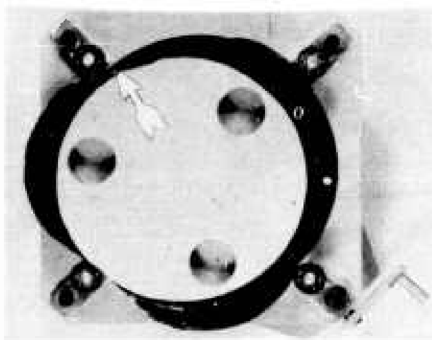
W. G. Photo No. L-9695 - File No. 50584

The model shown is a three lobe cycloidal cam reducer. Position 1 shows the relation of cam to input disk. The input disk (dot) rotates in the counterclockwise direction while the cam (arrow) moves in the clockwise direction.



W. G. Photo No. L-9697 - File No. 50584

The model shown is a three lobe cycloidal cam reducer. Position 2 shows input disk rotated once in counterclockwise direction while cam has moved one-third of a revolution in the clockwise direction.

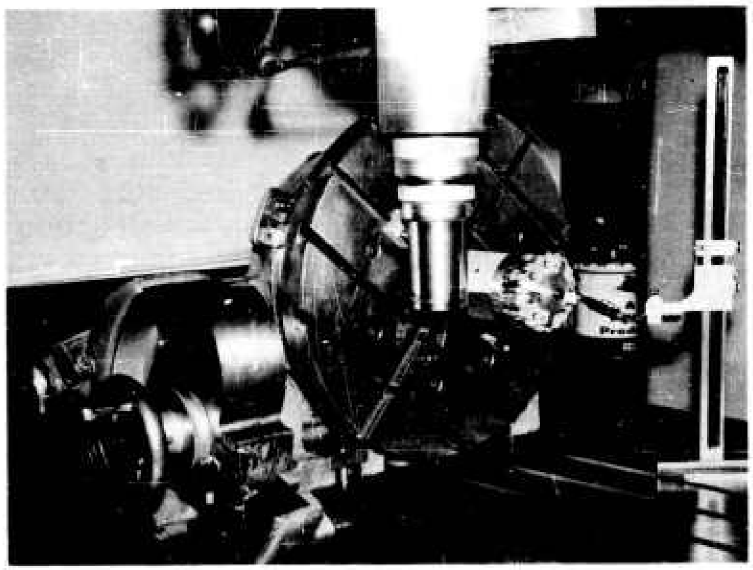


W. G. Photo No. L-9696 - File No. 50584

The model shown is a three lobe cycloidal cam reducer. Position 3 shows the input disk rotated once in the counterclockwise direction while the cam has moved two-thirds of a revolution in the clockwise direction.

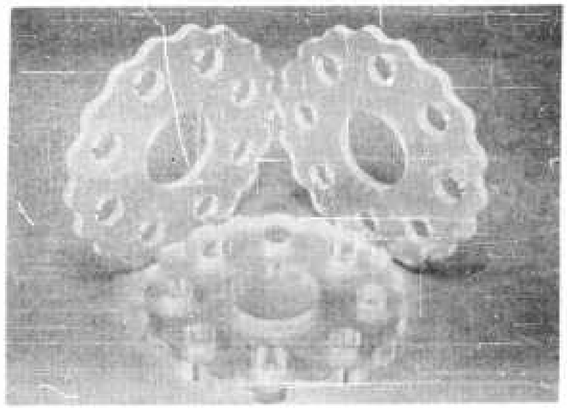
W. G. Photo No. L-9917 - File No. 50584

The tool shown is the fly-cutter used in conjunction with the Pratt & Whitney Jig Boring Machine, for machining the display model cam profiles.



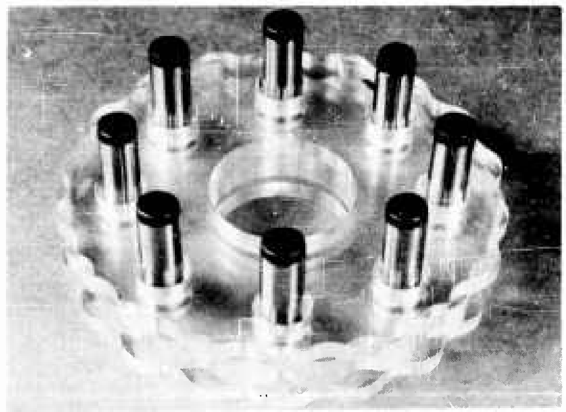
W. G. Photo No. L-9914 - File No. 50584

One set of three model cam plates with magnesium alloy backup plates mounted on the Pratt & Whitney 3E Jig Boring Machine for fly-cutting of cam lobes.



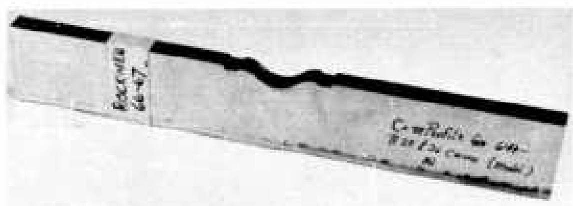
W. G. Photo No. L-9915 - File No. 50584

A set of three finished cam plates for the display model.



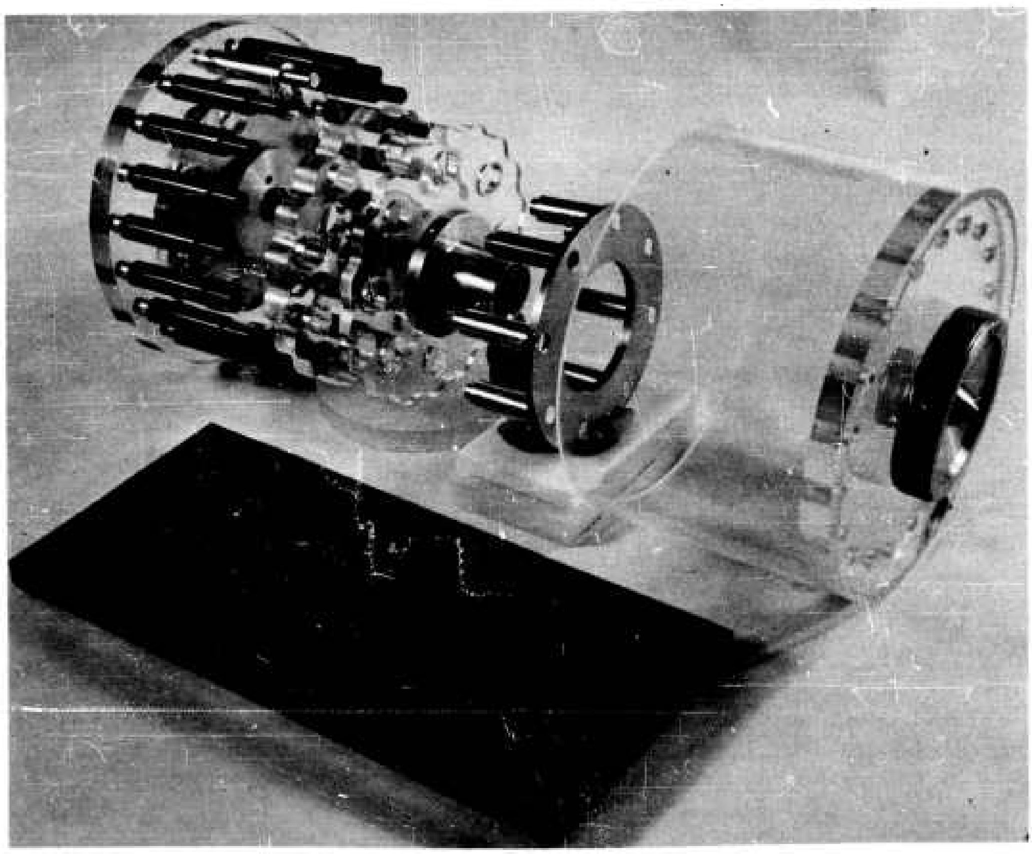
W. G. Photo No. L-9916 - File No. 50584

The set of cam plates for the display model phased together and located by the eight reverter pins.



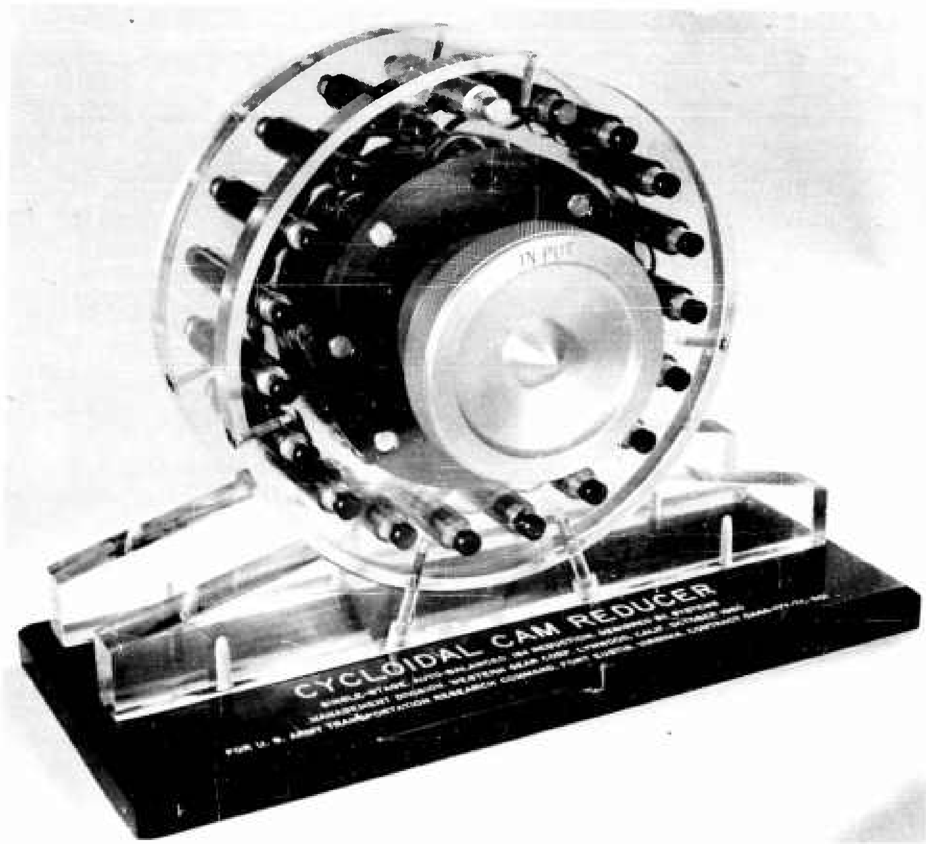
W. G. Photo No. L-10080 - File No. 50584

Cam Plate Template

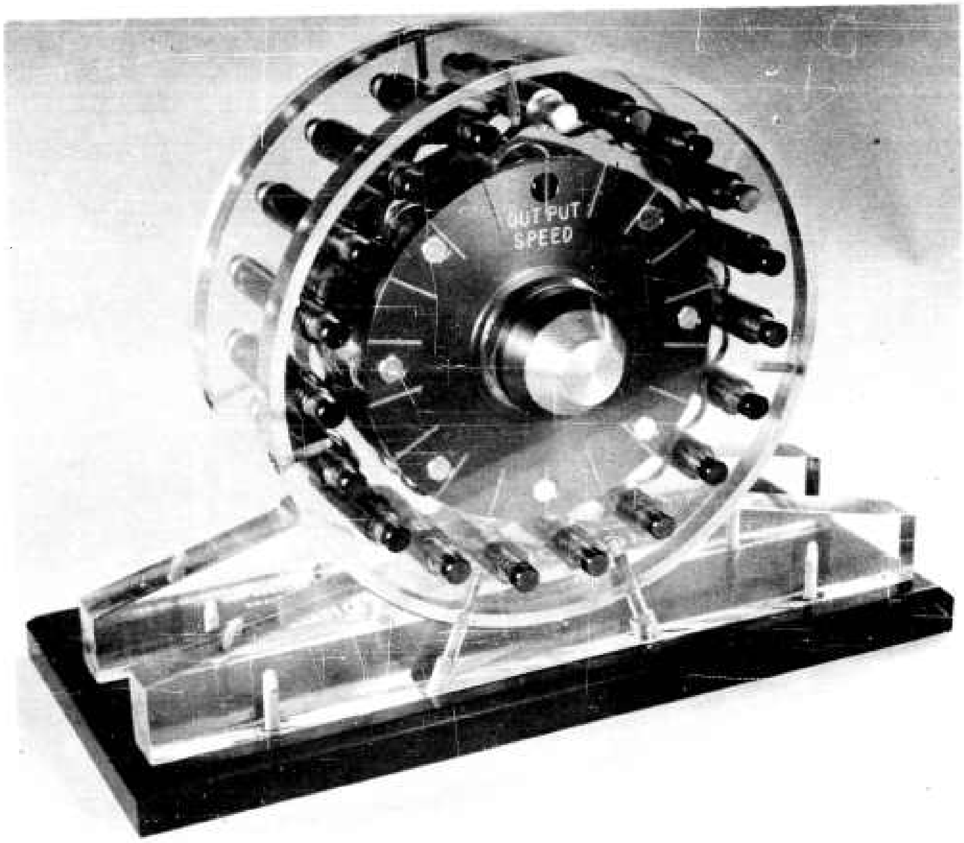


W. G. Photo No. L-9980 - File No. 50584

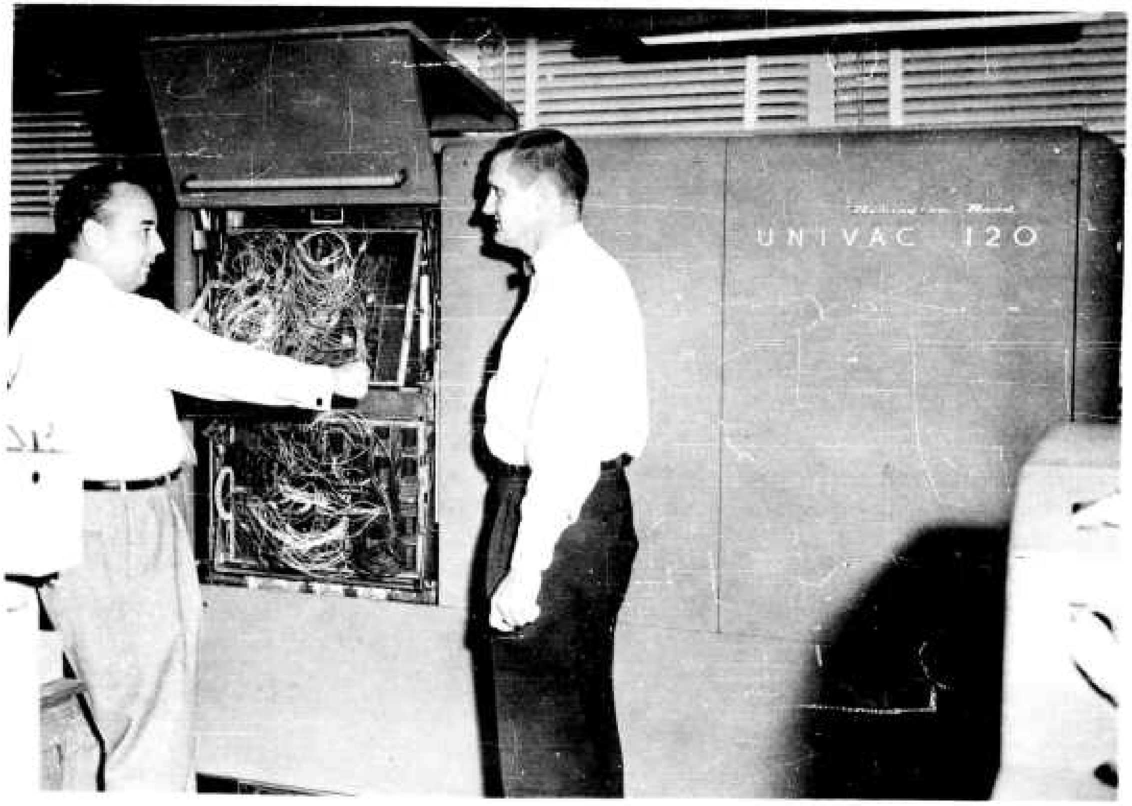
Exploded View of Display Model



W. G. Photo No. L-9996 - File No. 50584 Front View of Display Model



W. G. Photo No. L-9997 - File No. 50584 Back View of Display Model



W. G. Photo No. L-10112 - File No. 50584

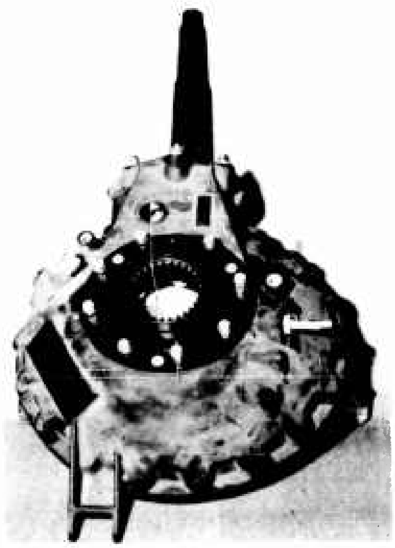
Computer Set-up for Cycloid Cam Co-ordinates

Helicopter

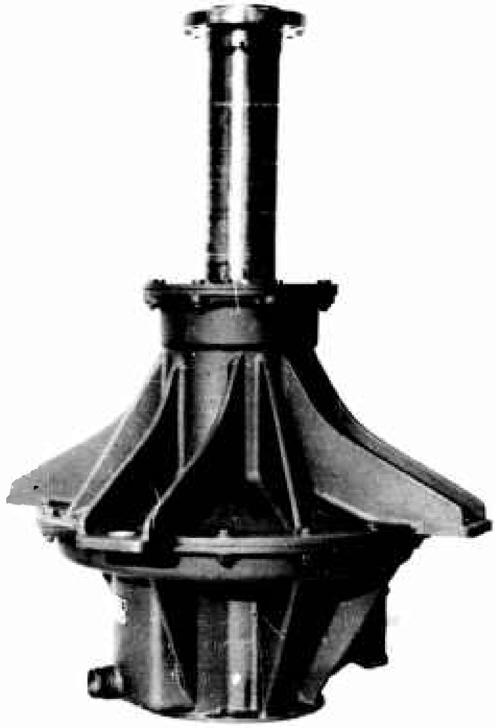
Main Rotor

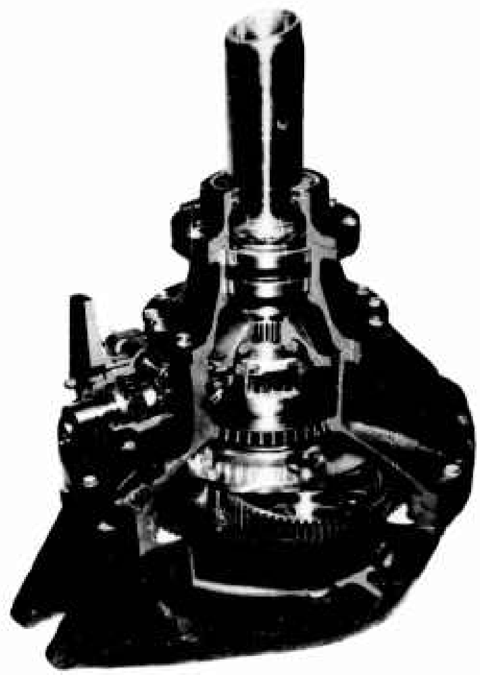
Drive

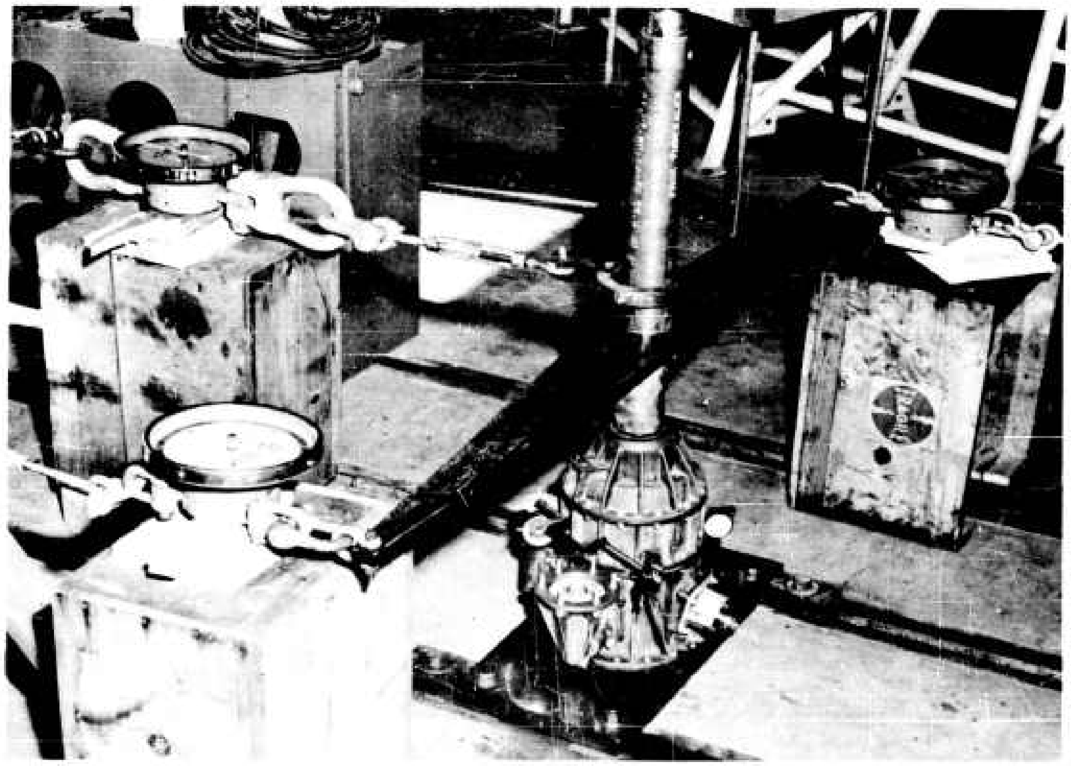
Transmissions



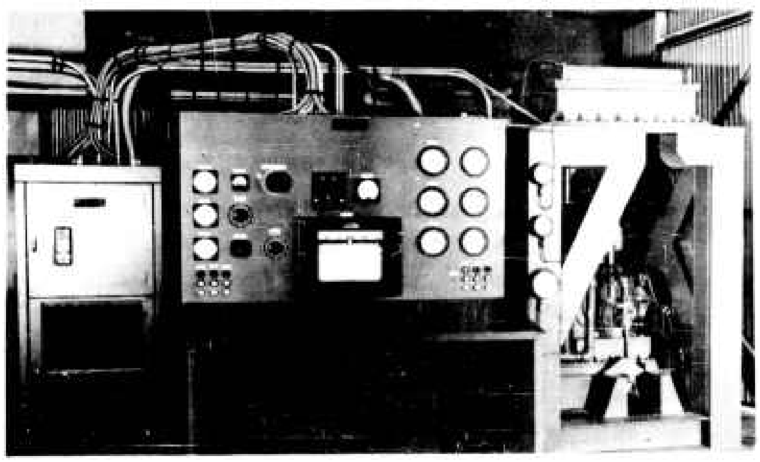
Helicopter
Main Rotor
Drive
Transmissions



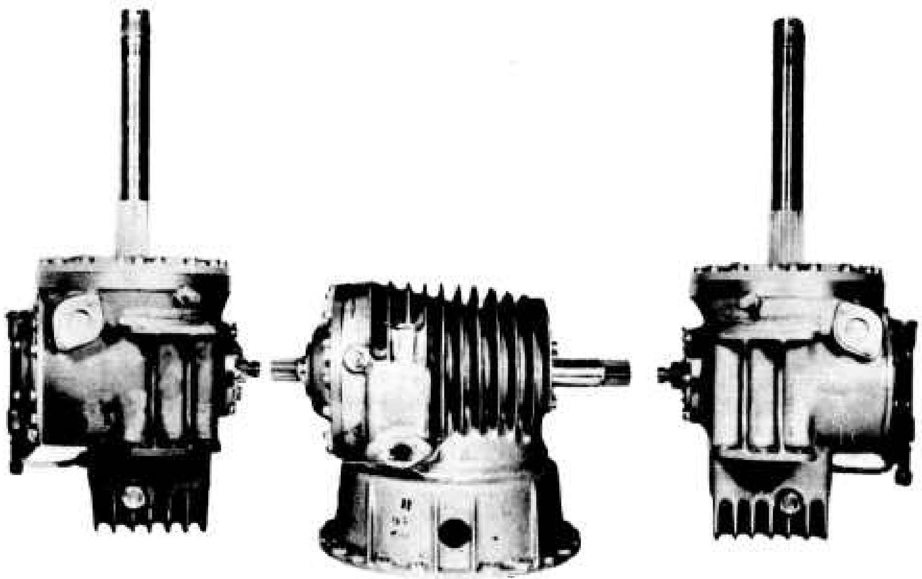


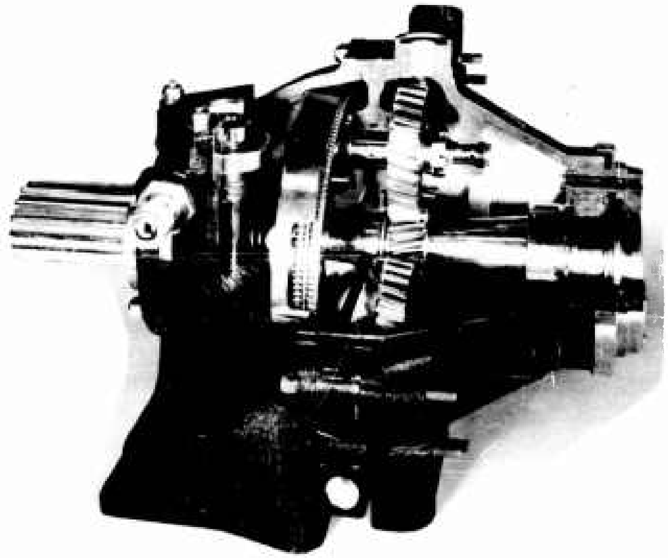


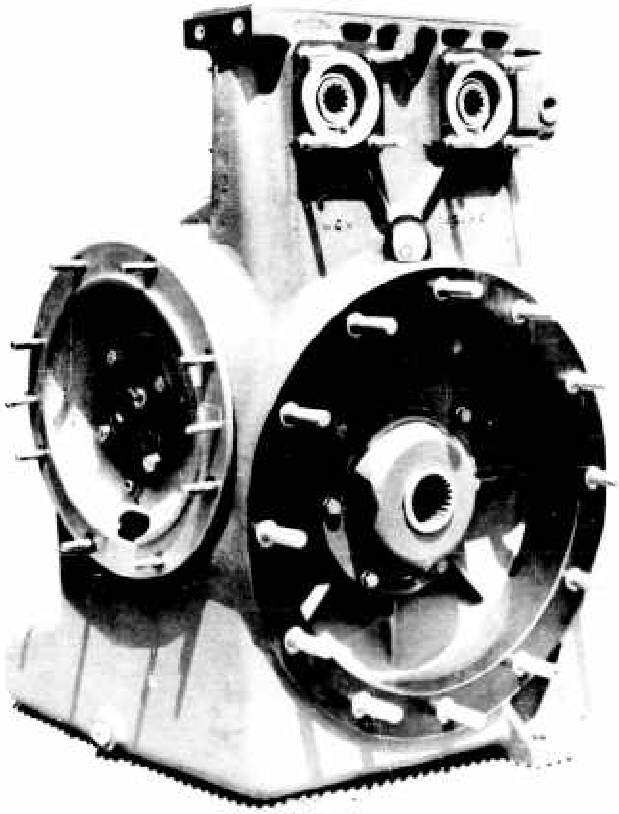
Helicopter Transmission Tests



VTOL Dual Rotor
Turbine Powered
Drive System

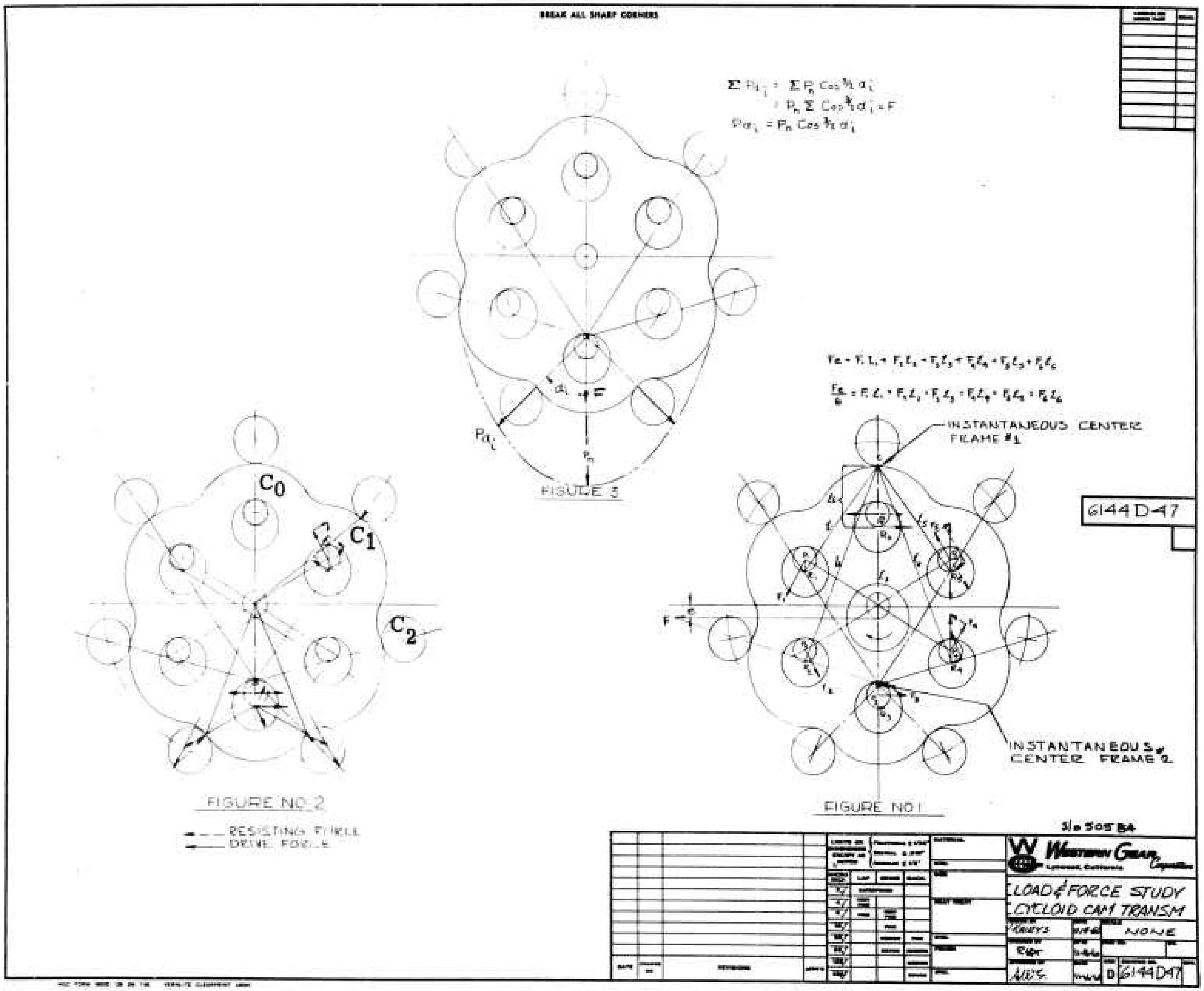






Turbine
Driven
Gearboxes

ILLUSTRATIONS


BREAK ALL SHARP CORNERS
FIGURE 3
FIGURE NO 2
RESISTING FORCE
DRIVE FORCE
FIGURE NO 1
INSTANTANEOUS CENTER FRAME #1
INSTANTANEOUS CENTER FRAME #2
6144D47
LOAD & FORCE STUDY
CYCLOID CAM TRANSM
NONE
D 6144D47

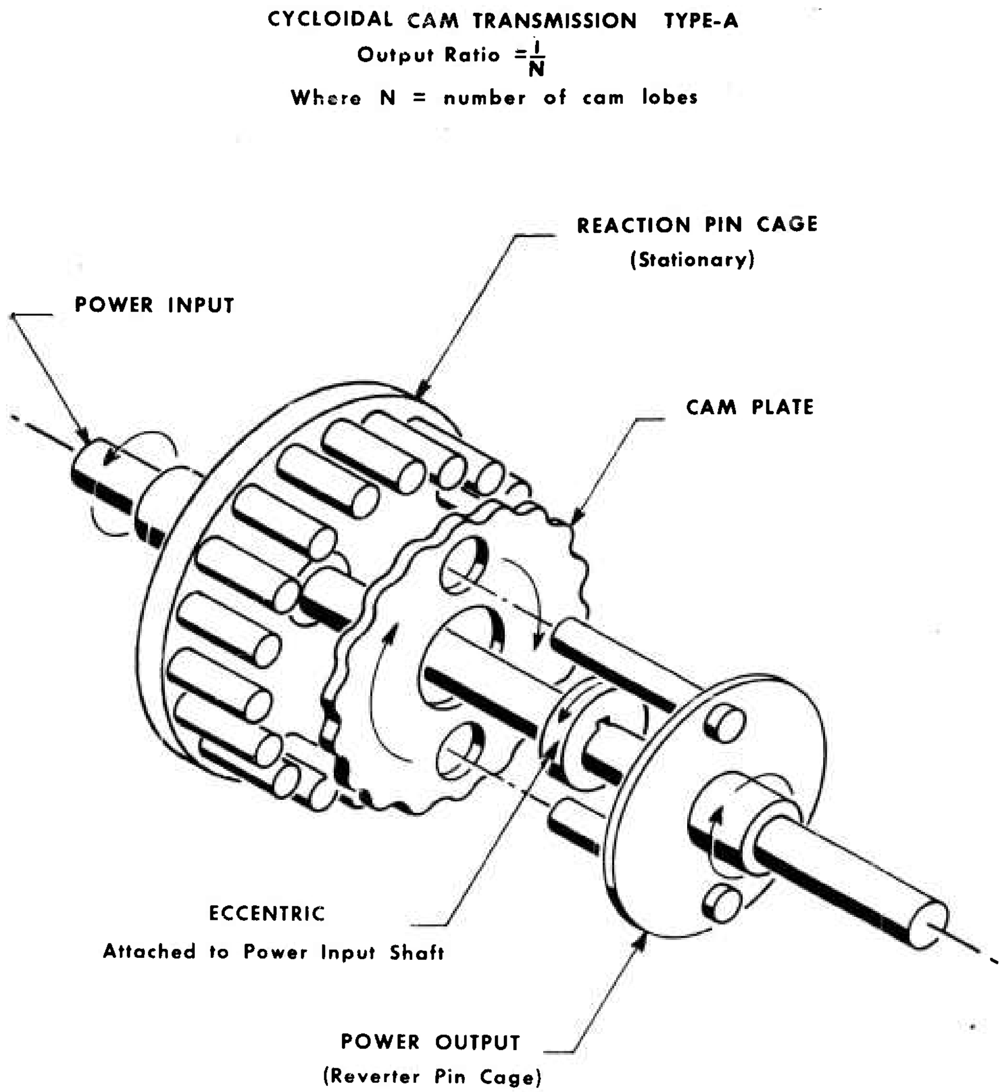


Figure 4. Cycloidal Cam Transmission - Type A

CYCLOIDAL CAM TRANSMISSION TYPE B

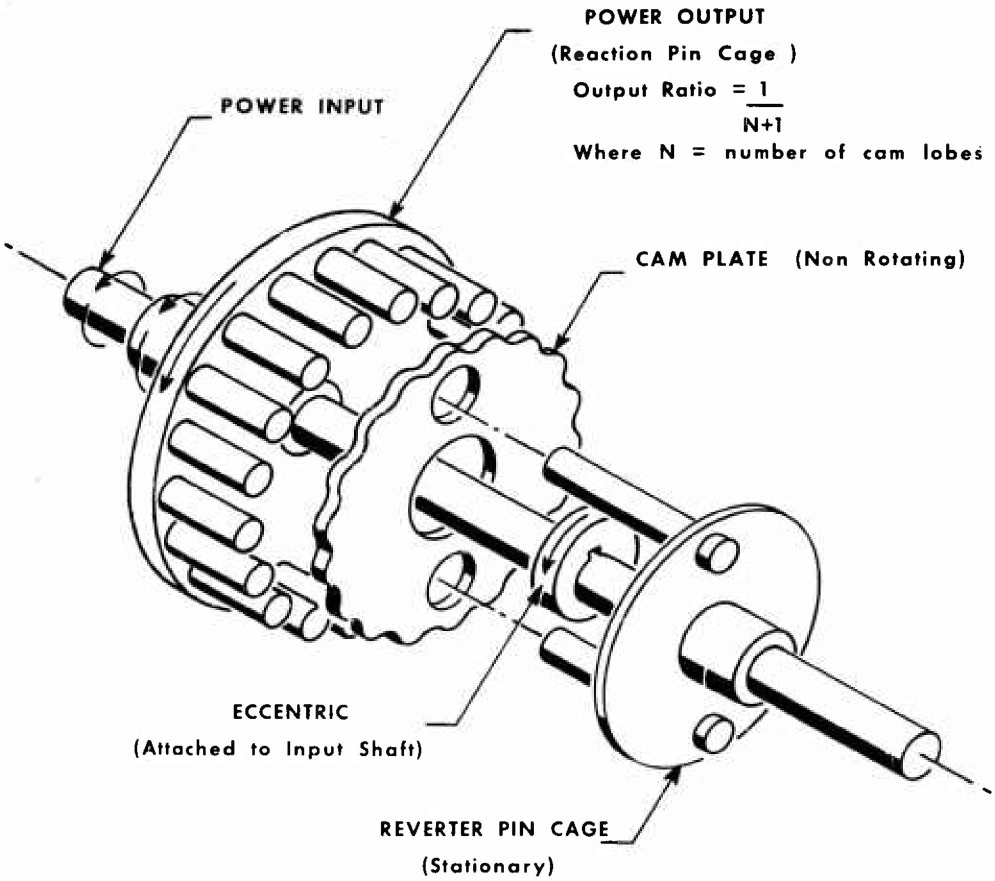


Figure 5. Cycloidal Cam Transmission - Type B

CYCLOIDAL CAM TRANSMISSION TYPE C

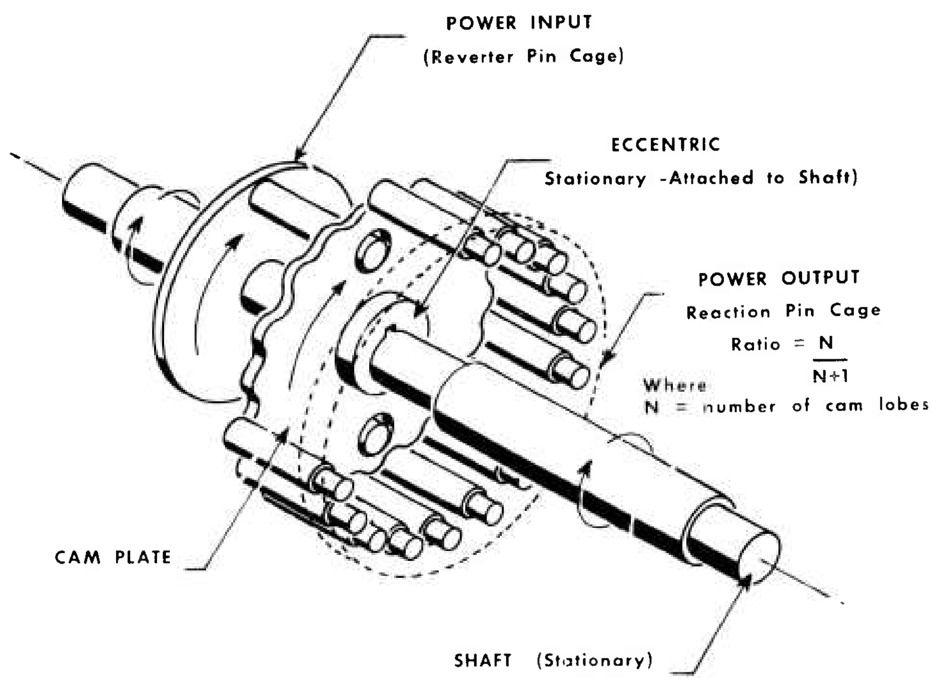


Figure 6. Cycloidal Cam Transmission - Type C

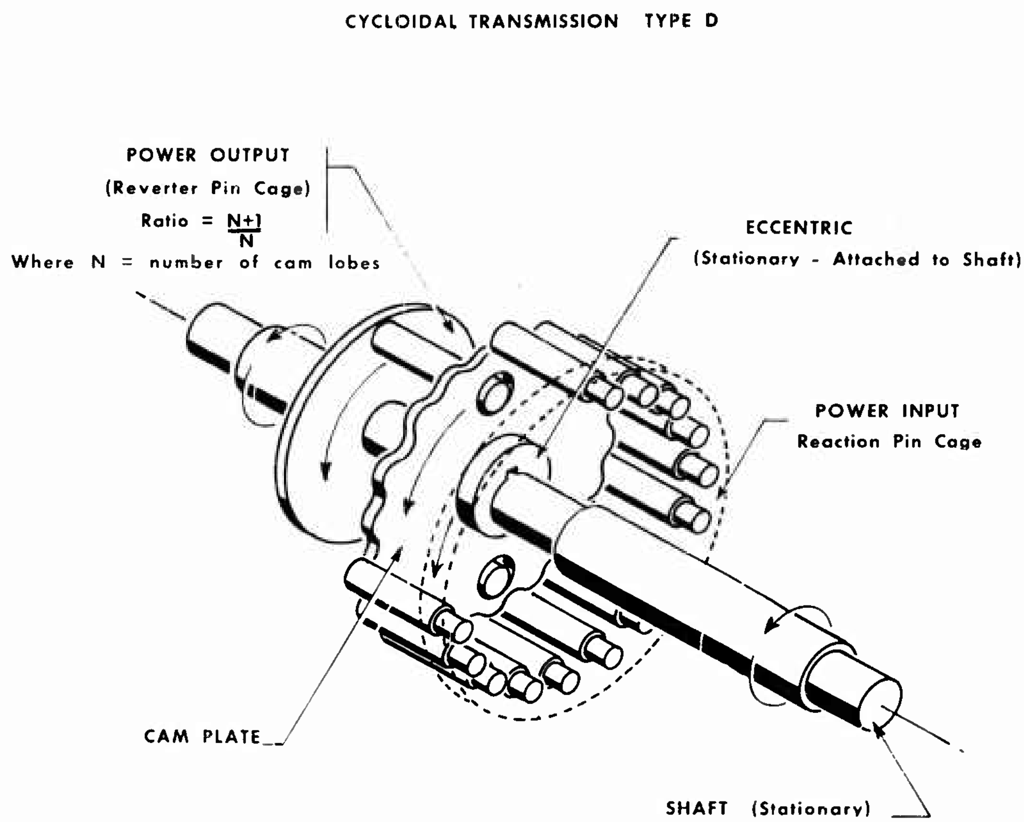


Figure 7. Cycloidal Cam Transmission - Type D

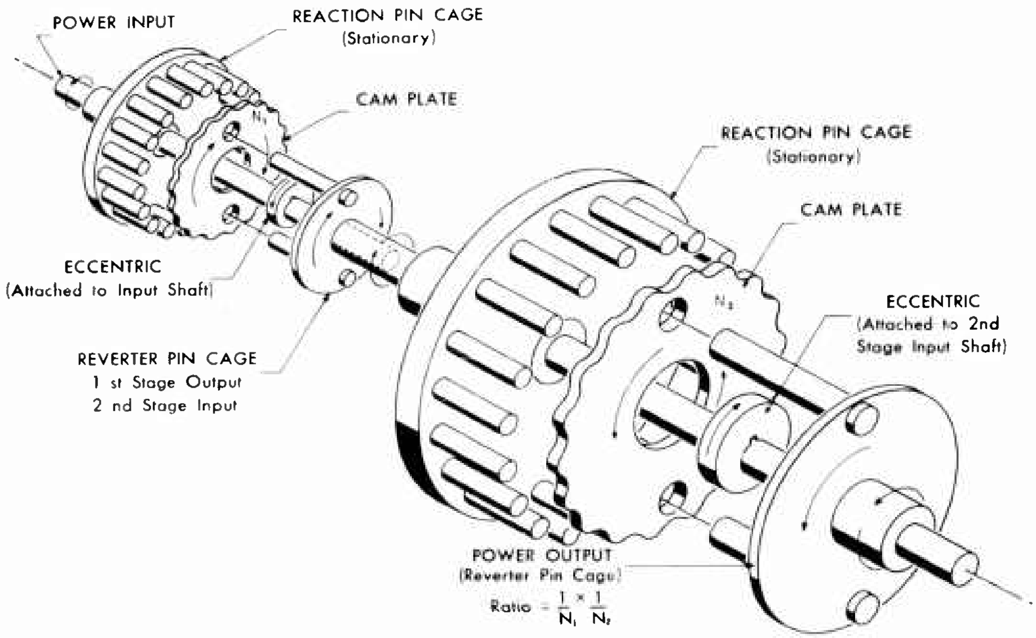


Figure 8. Two Stage A to A Arrangement

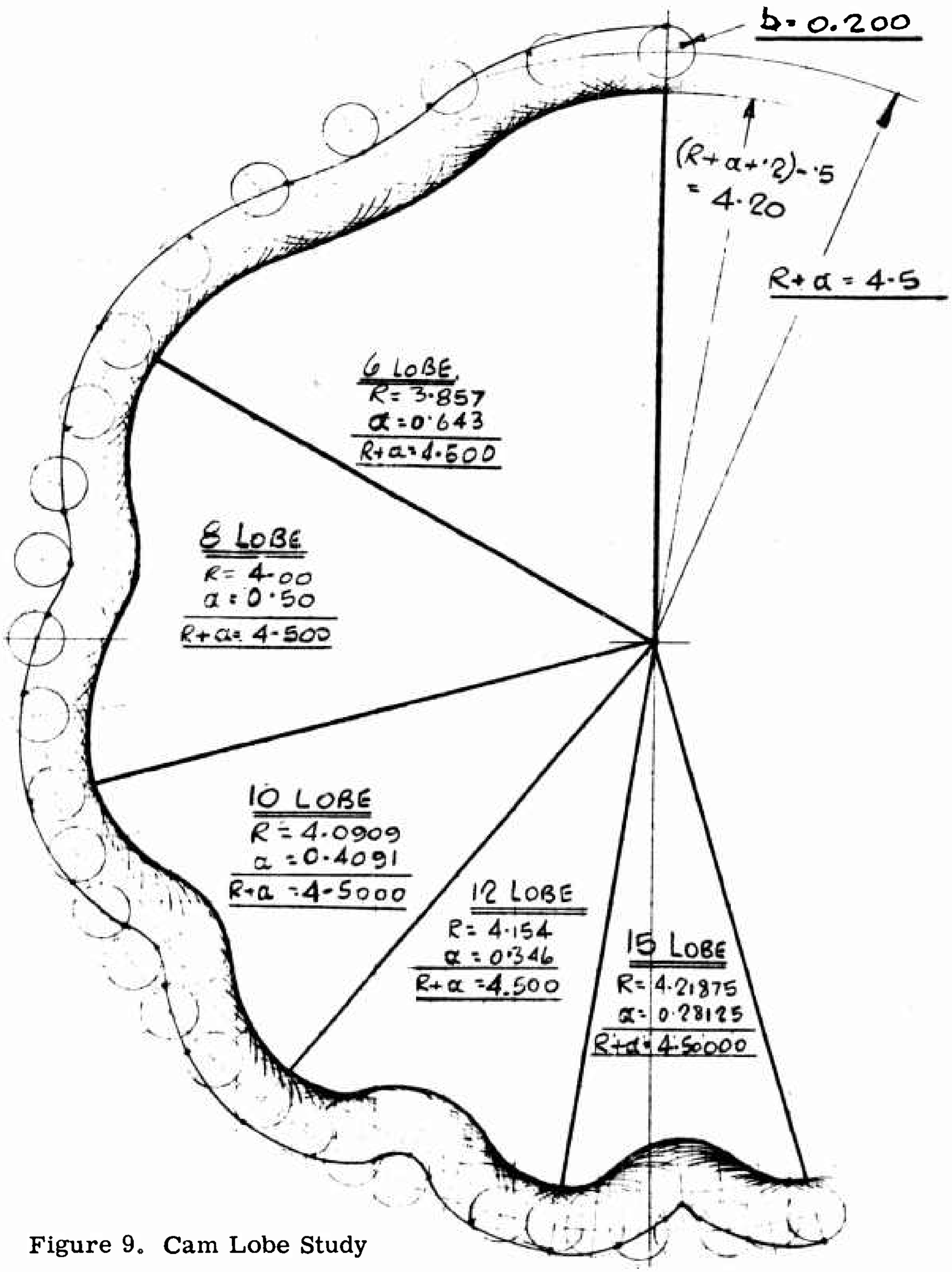


Figure 9. Cam Lobe Study

CYCLOIDAL CAM TRANSMISSION

The unit shown is an 18:1 reducer. The speed reducing effect is produced by rotating the input shaft inside the cam plates. Eccentrics on the input shaft force the cam lobes (which are of an epitrochoidal shape) against the fixed pins with a wedging action. This action turns the cams on their own centers by the amount of one lobe pitch for each revolution of the input. The reverting pins and holes translate the cam revolutions to the output cage with uniform velocity. Three cam plates are used to accomplish dynamic balance of the system.

The operation is extremely quiet, efficient and smooth, with no appreciable backlash.

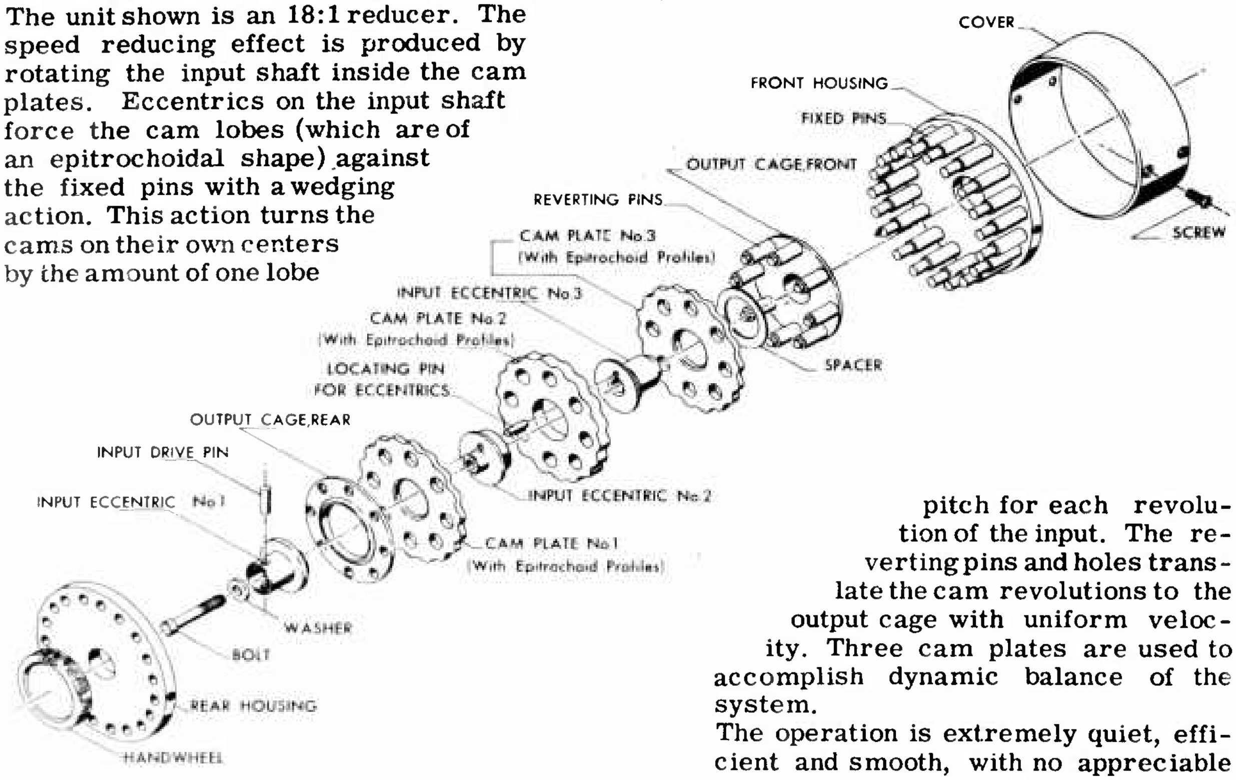


Figure 10 Exploded View of Display Model

$$X = (R+a)\cdot\sin\phi + b\sin\left[\left(\frac{R+a}{a}\right)\phi\right] - \rho\sin\left[\frac{\tan^{-1}\left(b\cdot\sin\left\{\frac{R}{a}\cdot\phi\right\}-\phi\right)}{a+b\cdot\cos\left\{\frac{R}{a}\cdot\phi\right\}}\right],\quad Y = (R+a)\cos\phi + b\cos\left[\left(\frac{R+a}{a}\right)\cdot\phi\right] - \rho\cos\left[\frac{\tan^{-1}\left(b\sin\left\{\frac{R}{a}\cdot\phi\right\}\right. \left.+\phi\right)}{a+b\cos\left\{\frac{R}{a}\cdot\phi\right\}}\right]$$

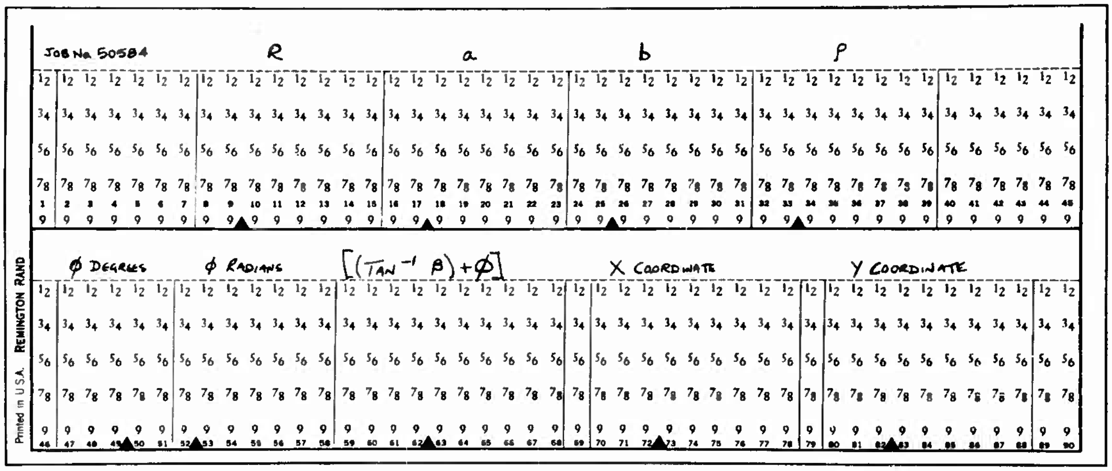


FIGURE 11 TYPICAL PROGRAMMING CARD FOR THE X & Y CO-ORDINATES OF THE CYCLOIDAL CAM PROFILES.

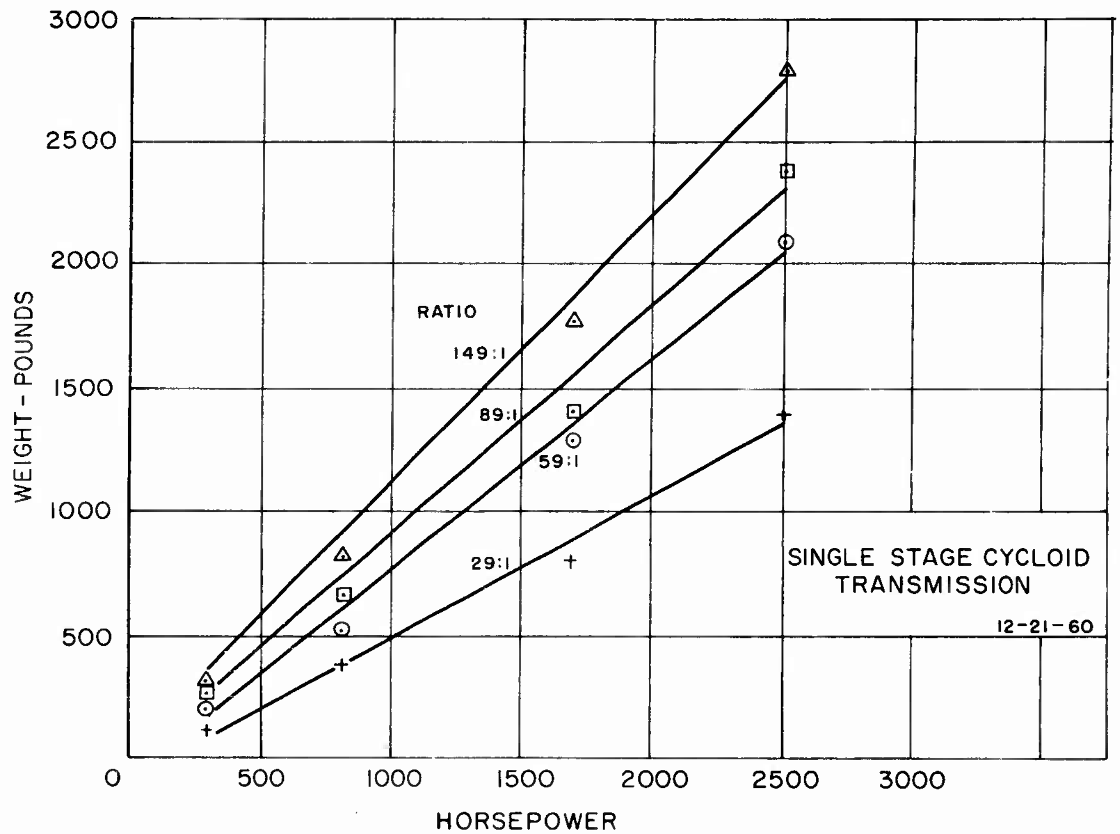


Figure 12. Single Stage Cycloid Transmission Weight vs Horsepower

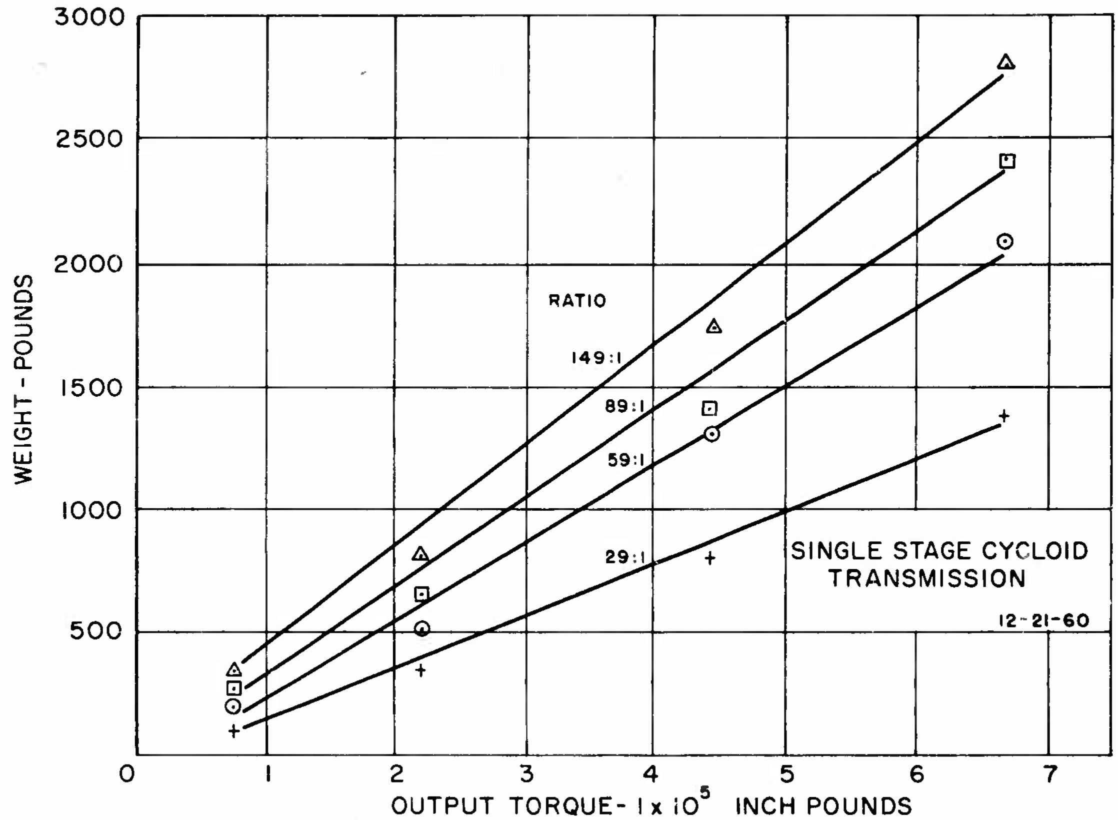


Figure 13. Single Stage Cycloid Transmission Weight vs Output Torque

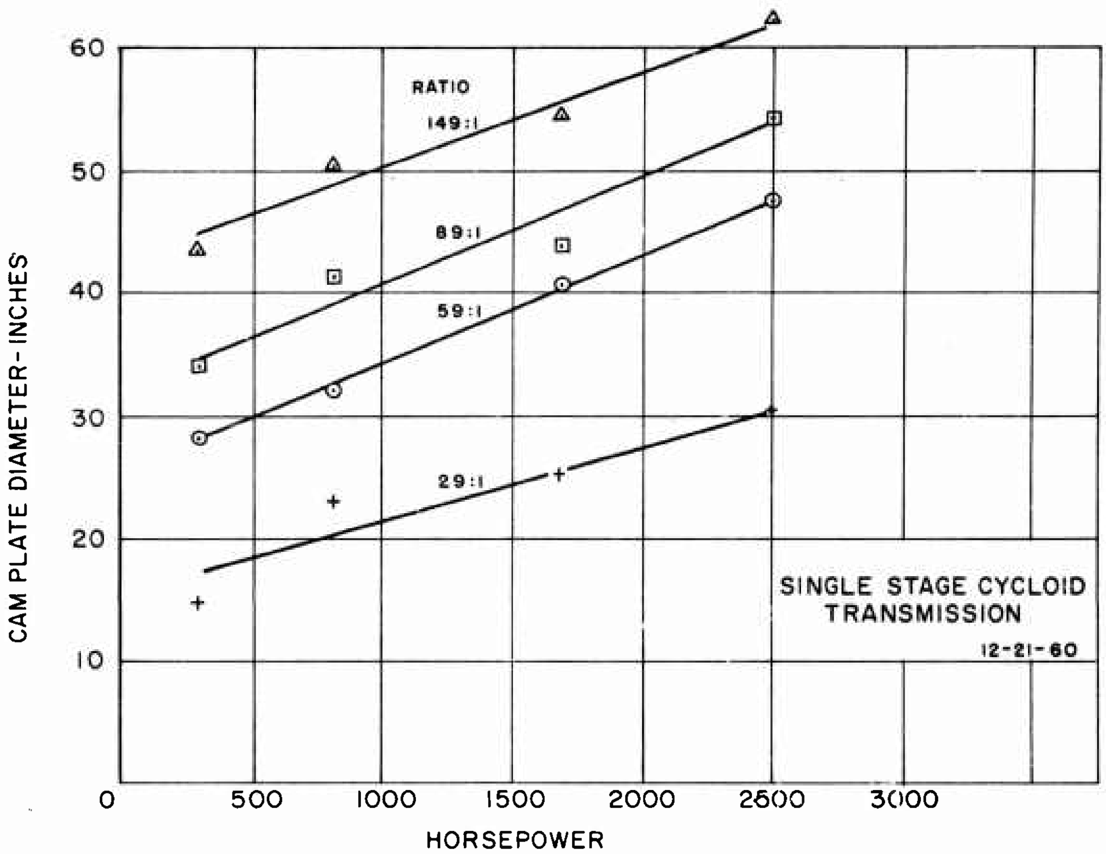


Figure 14. Single Stage Cycloid Transmission
Cam Plate Diameter vs Horsepower

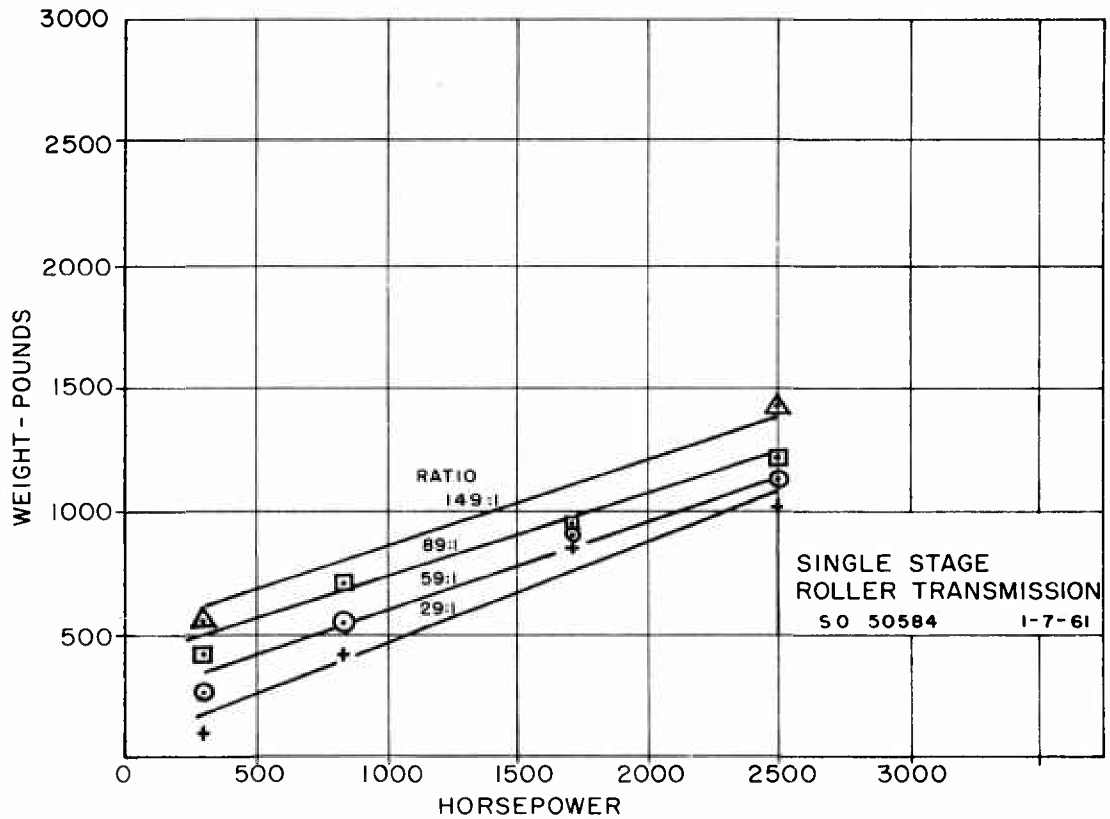


Figure 15. Single Stage Roller Transmission Weight vs Horsepower

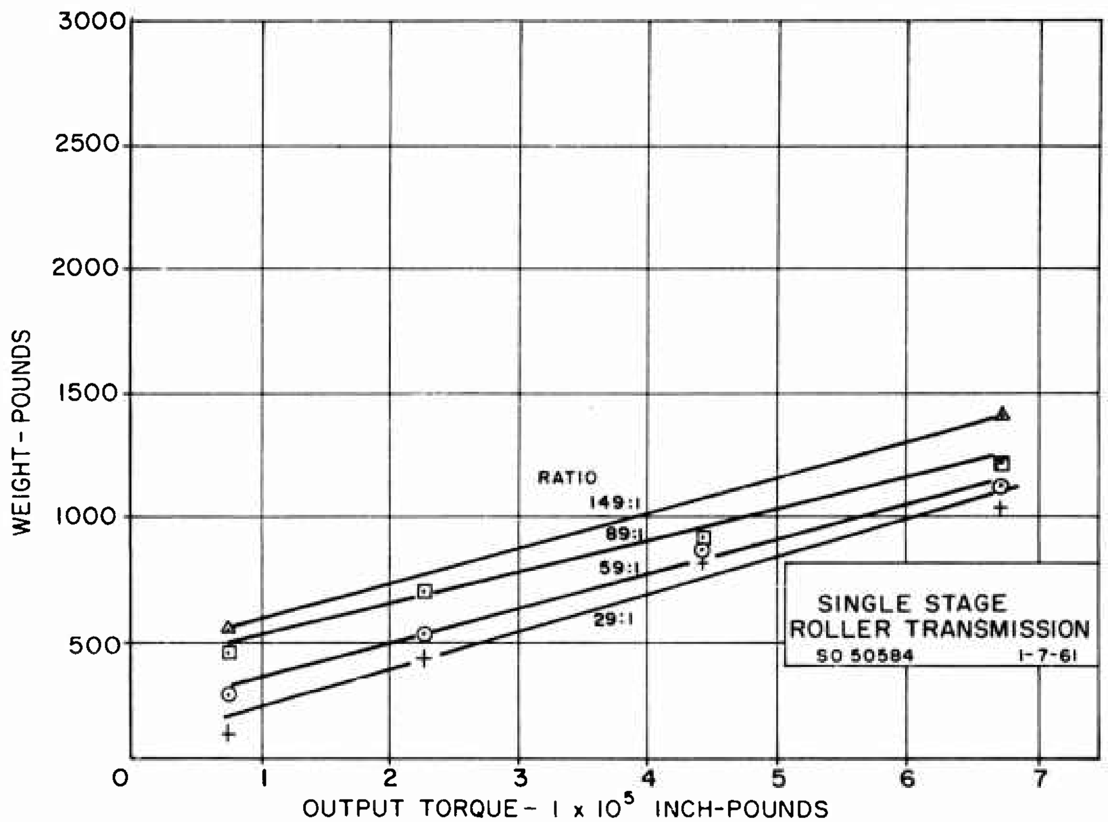


Figure 16. Single Stage Roller Transmission
Weight vs Output Torque

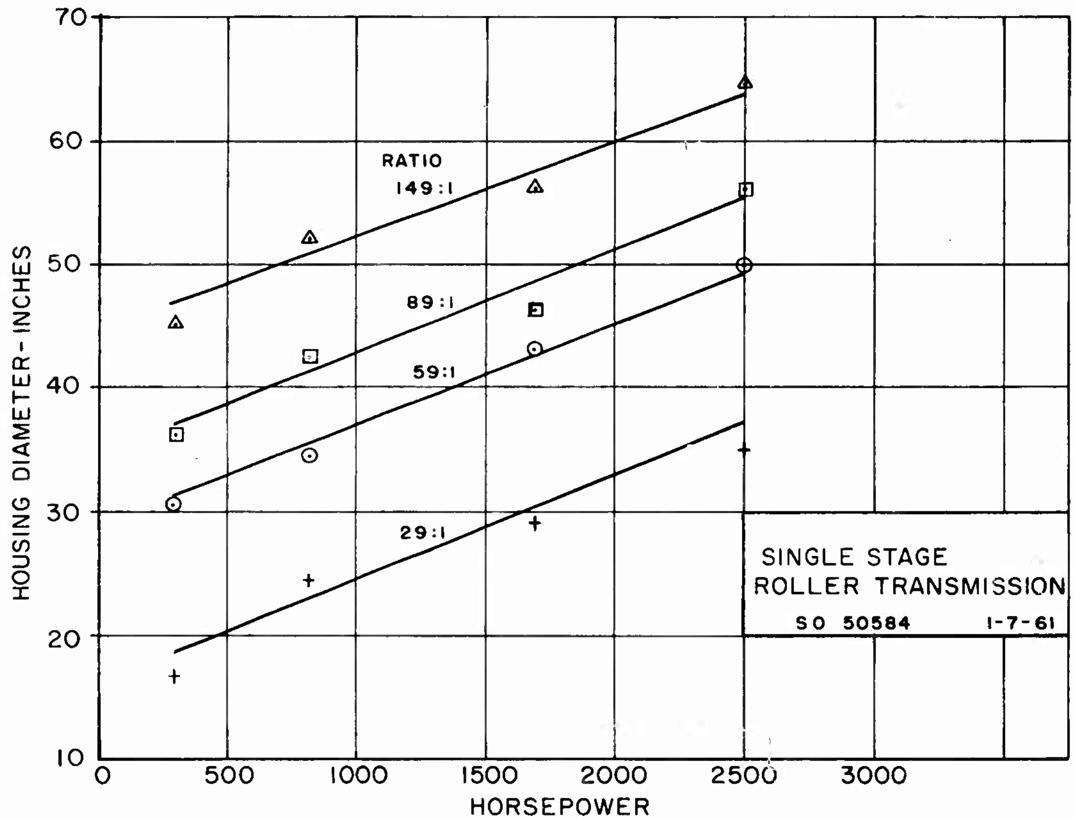


Figure 17. Single Stage Roller Transmission
Housing Diameter vs Horsepower

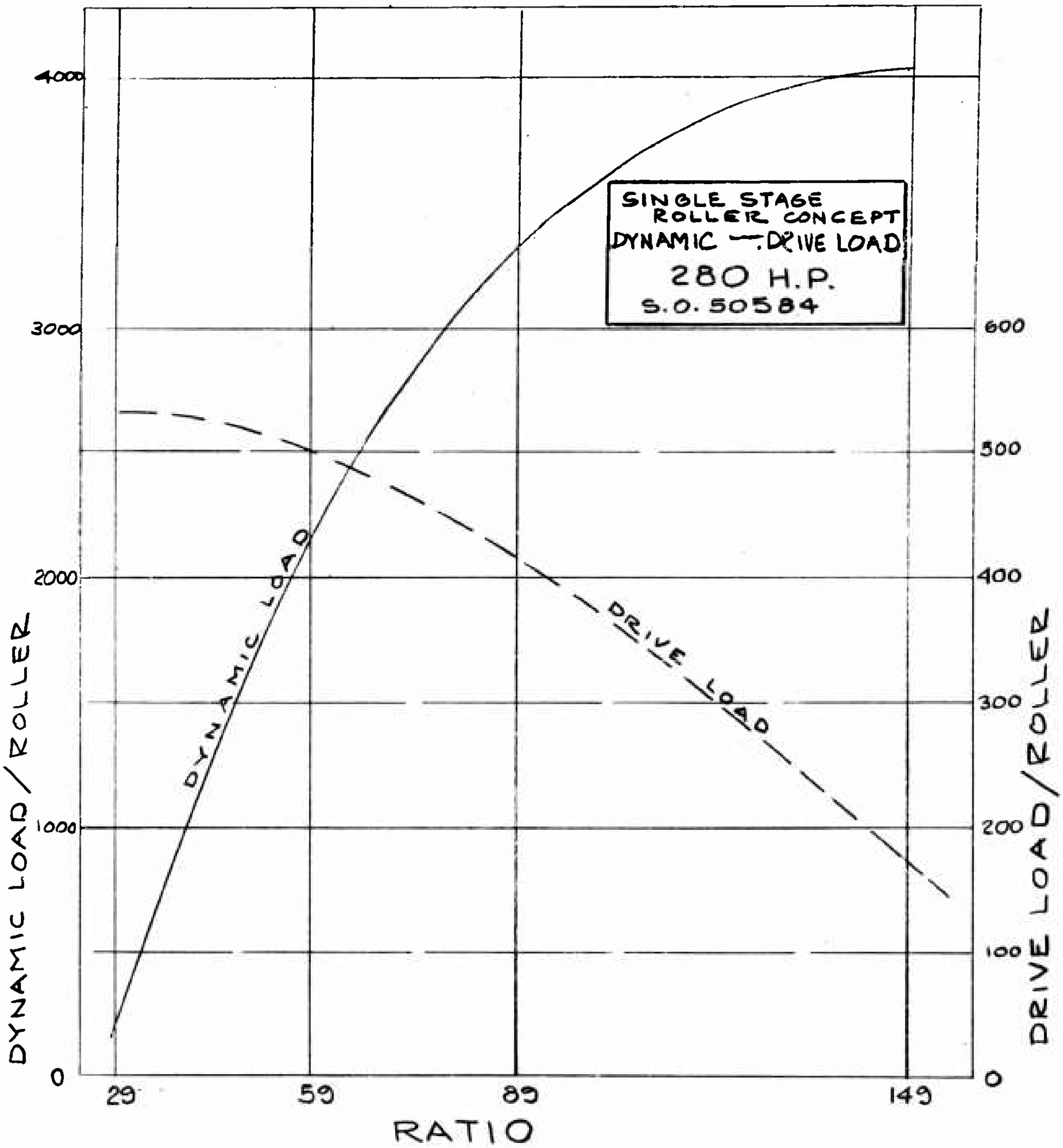


Figure 18. Single Stage Roller Concept
Dynamic-Drive Loads vs Ratio (280 HP)

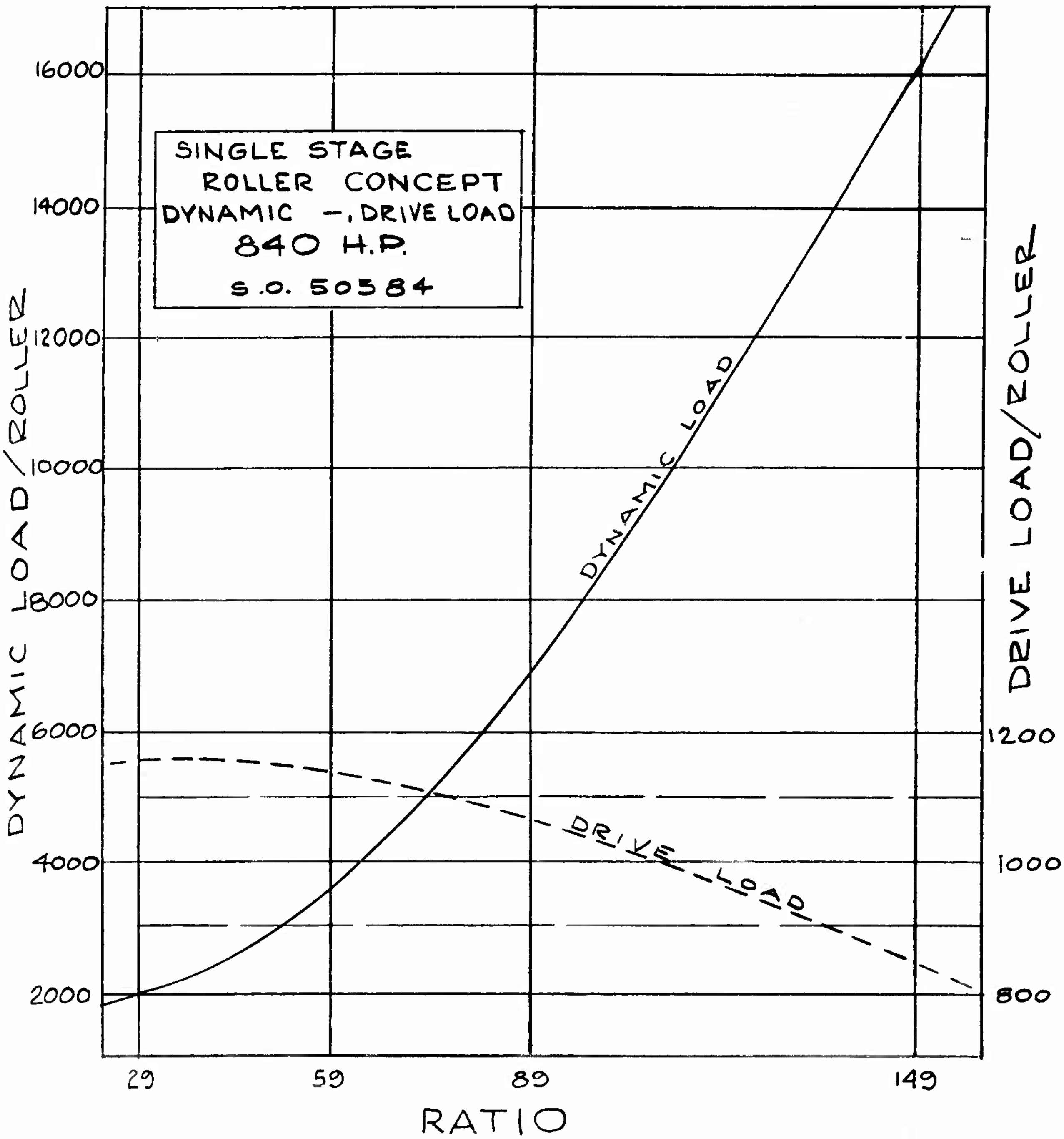


Figure 19. Single Stage Roller Concept
Dynamic-Drive Loads vs Ratio (840 HP)

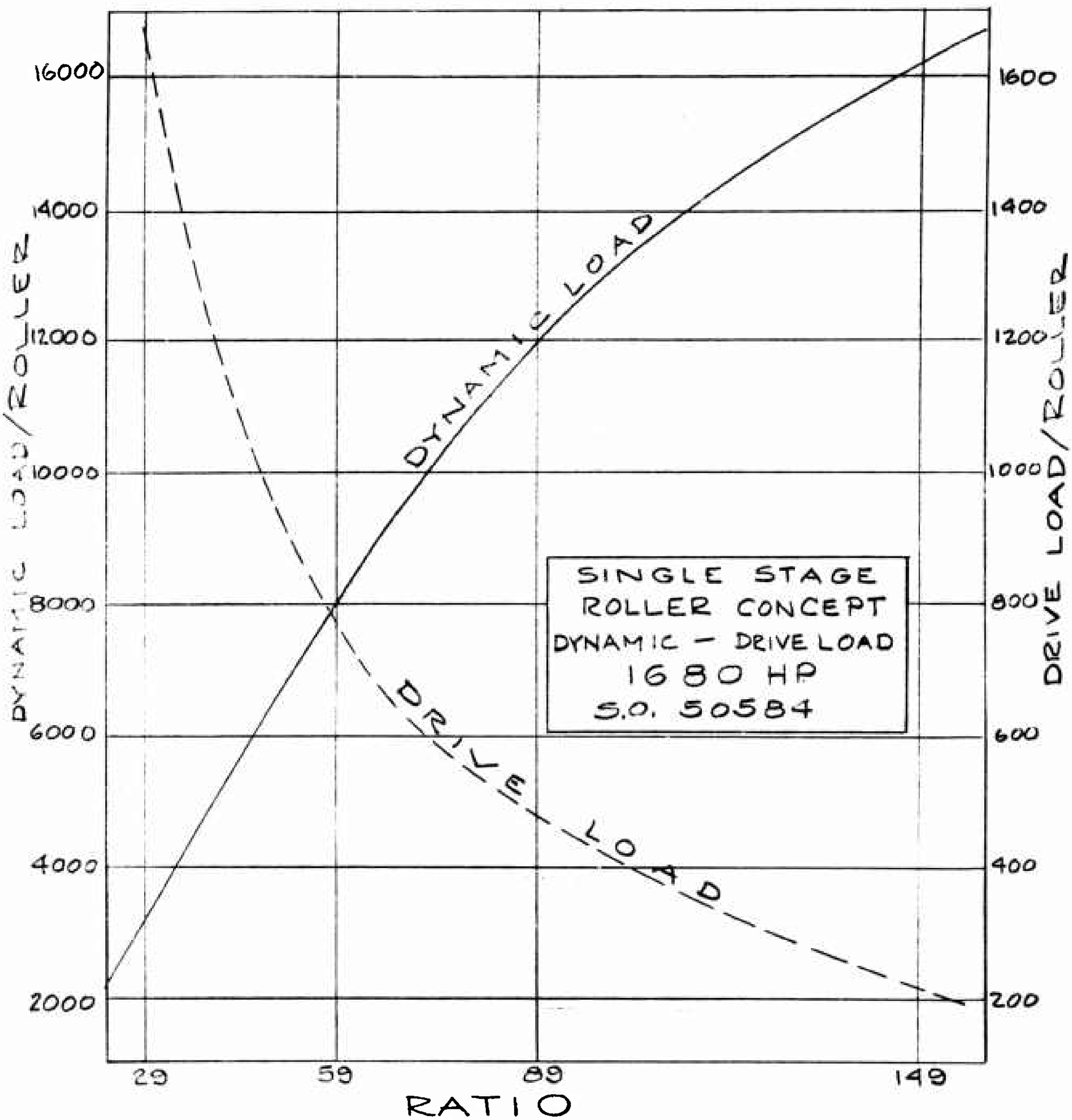


Figure 20. Single Stage Roller Concept
Dynamic-Drive Loads vs Ratio (1680 HP)

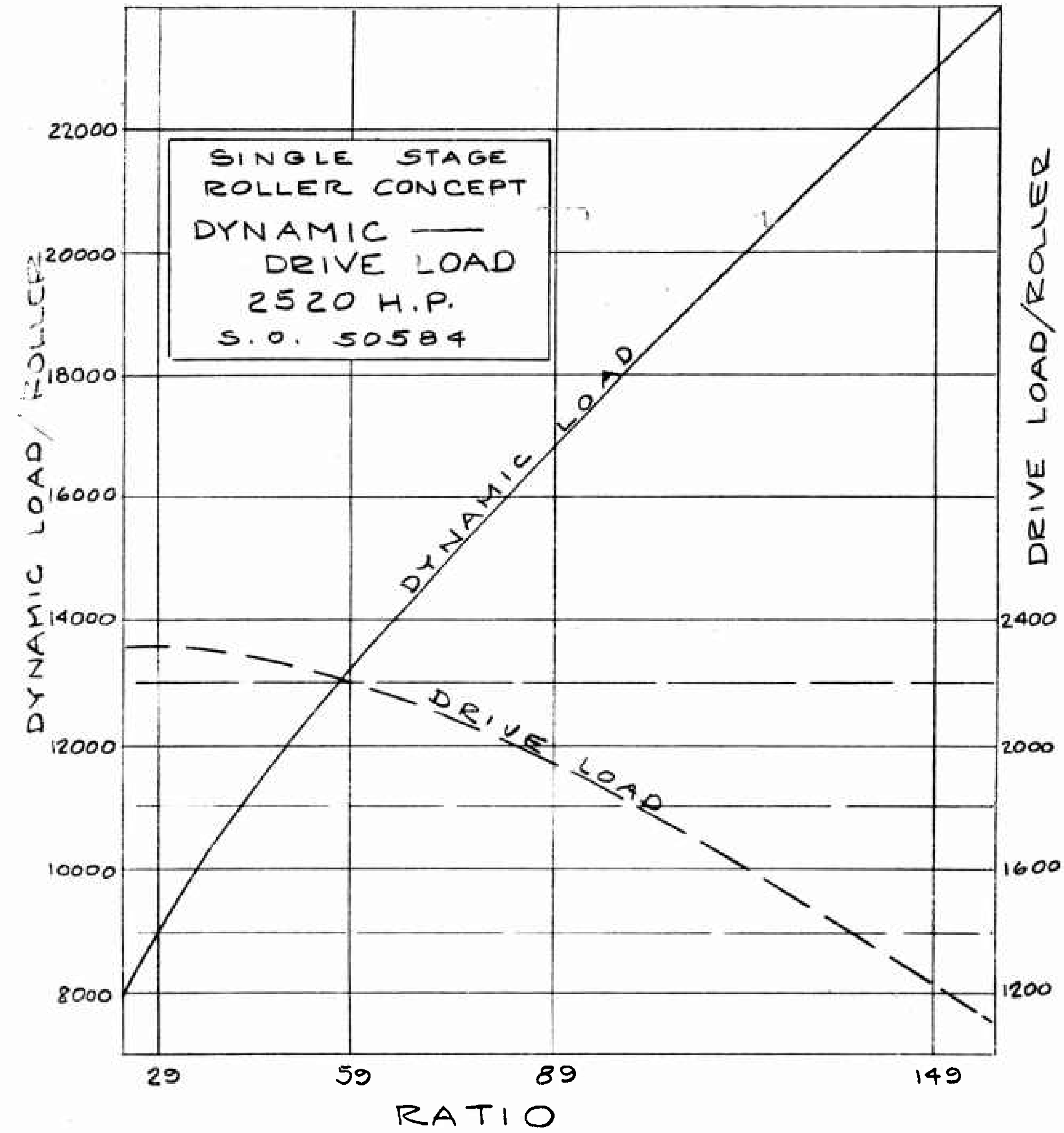


Figure 21. Single Stage Roller Concept
Dynamic-Drive Loads vs Ratio (2520 HP)

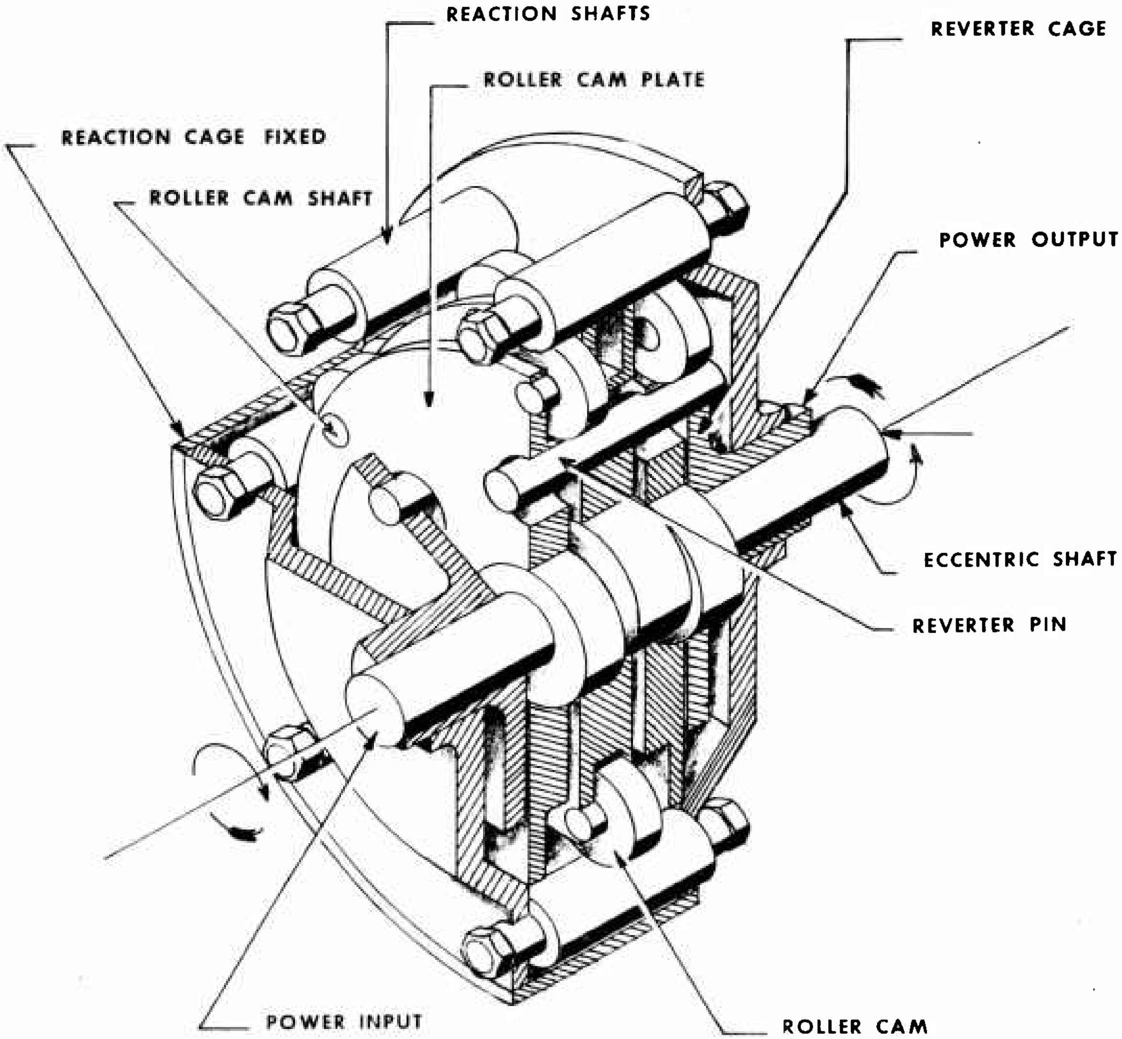
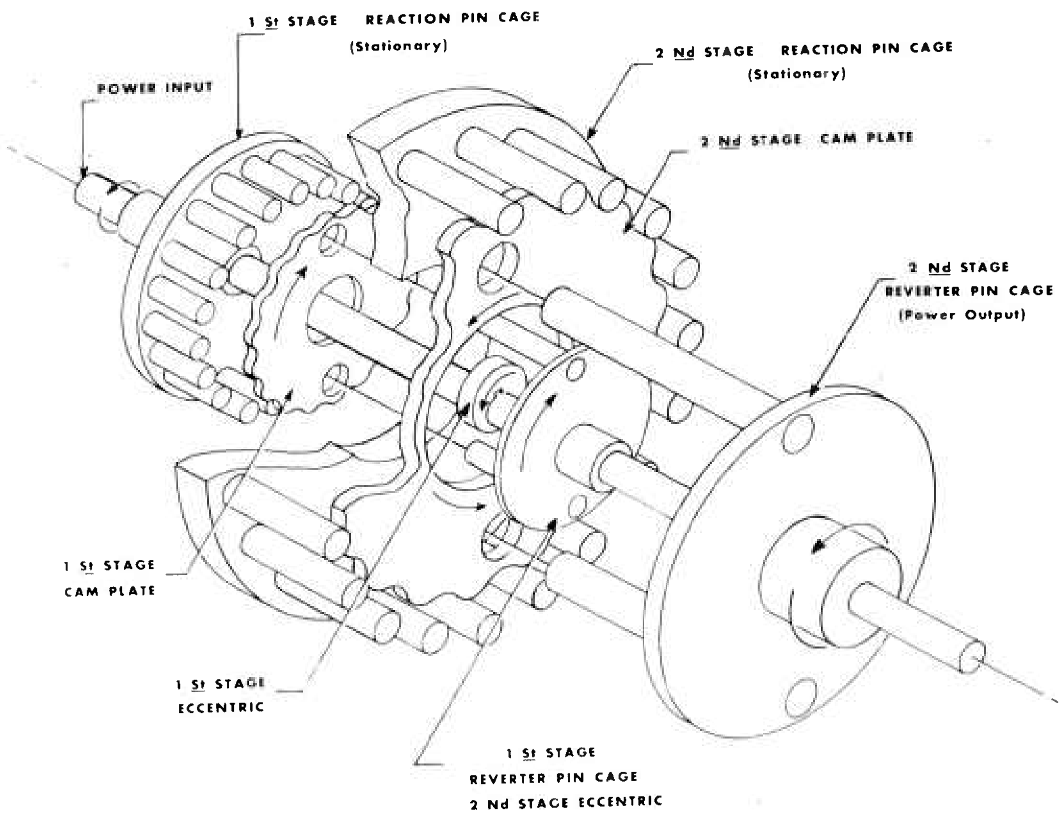


Figure 22. Single Stage Roller Cam Transmission



Cycloidal Cam Transmission 2 Stage—Telescopic Arrangement

Figure 23.

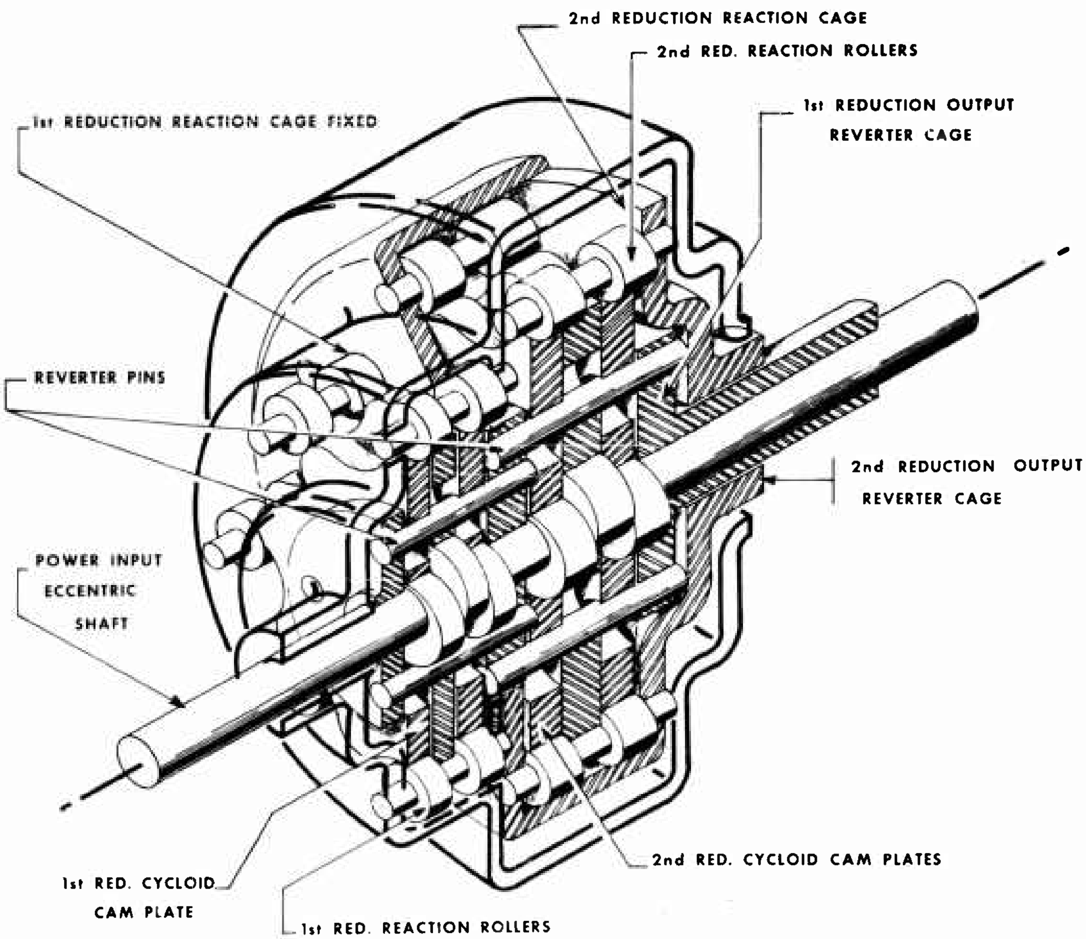


Figure 24. Single Stage Differential Cam Transmission

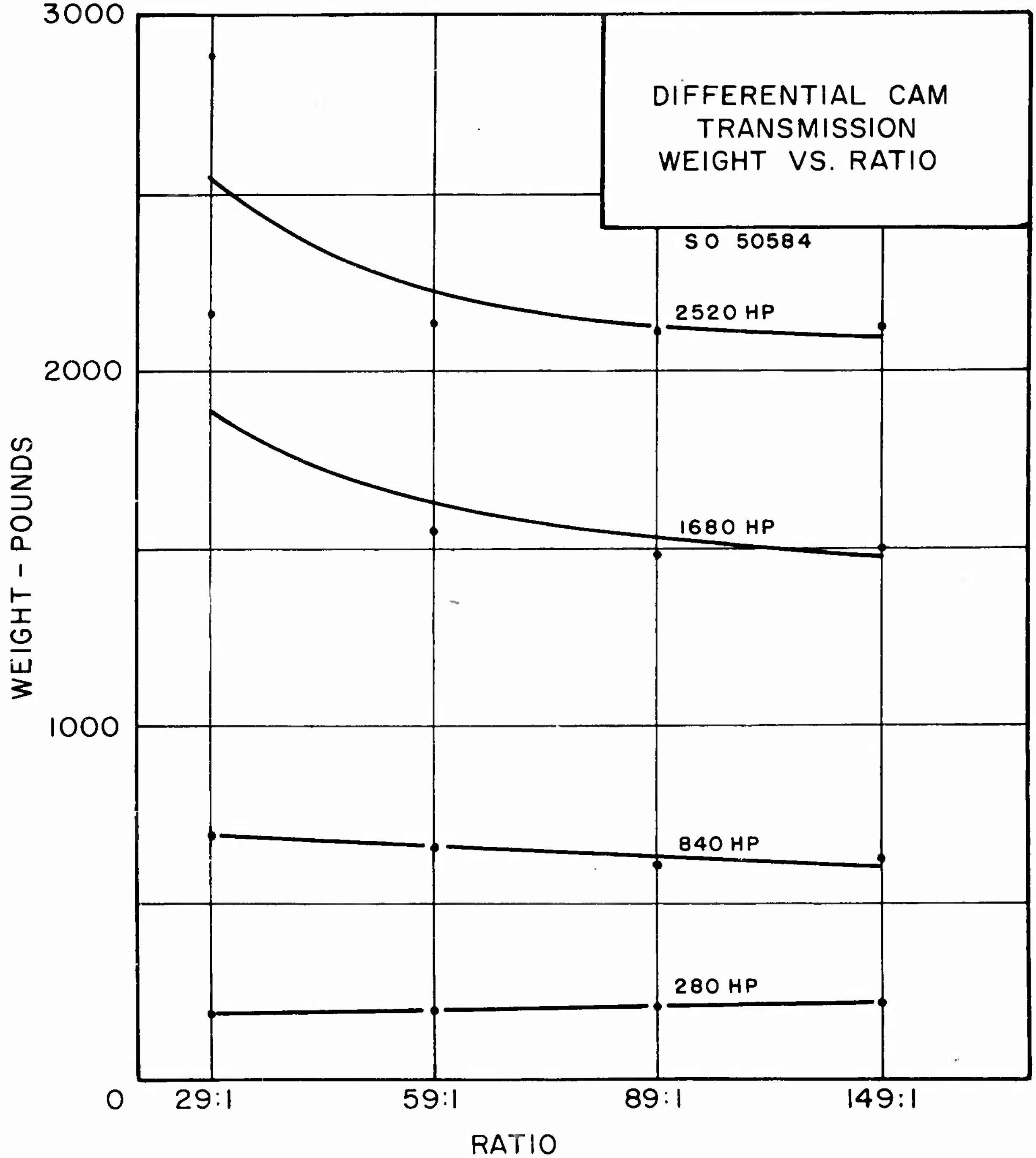


Figure 25. Differential Cam Transmission Weight vs Ratio

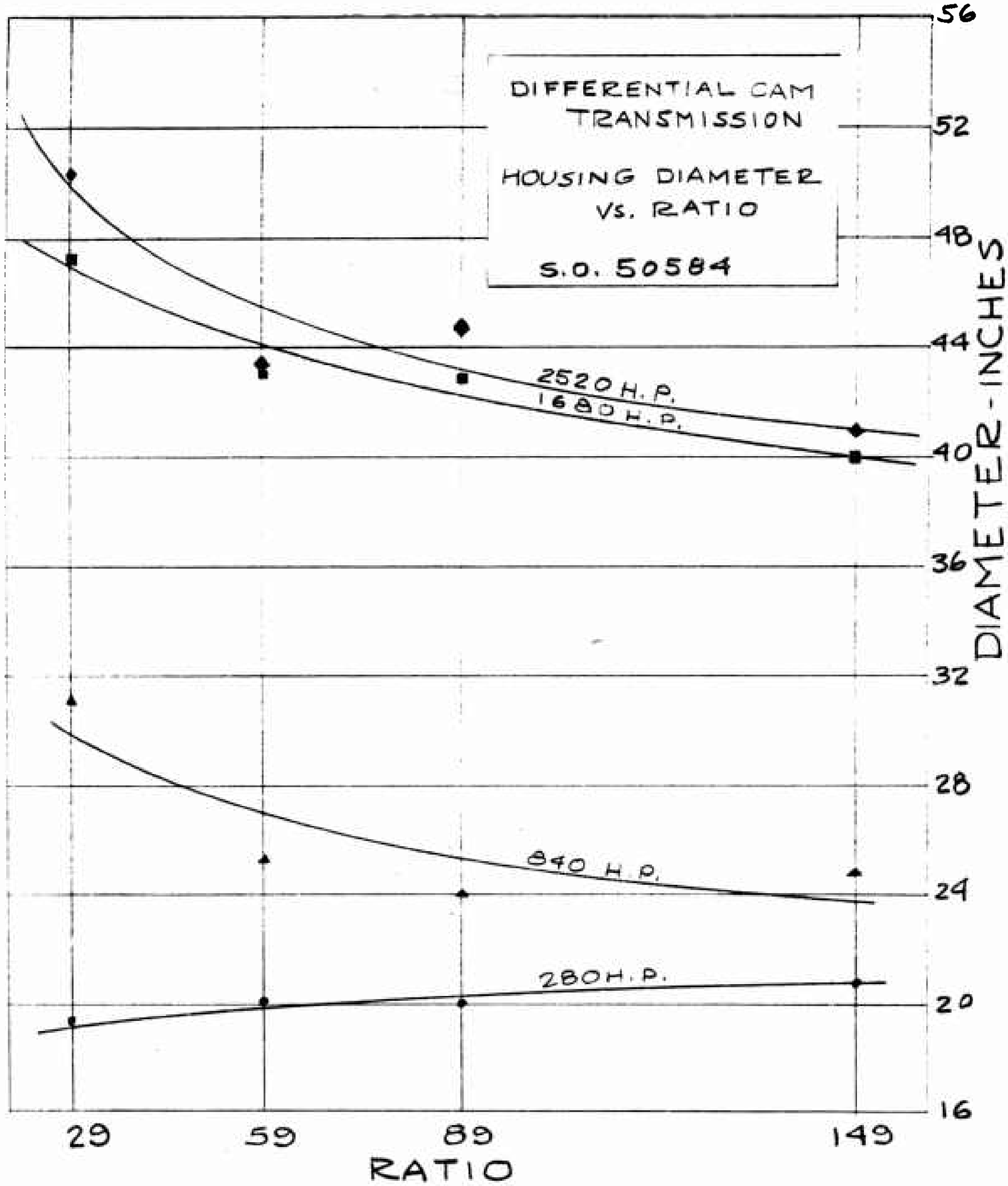
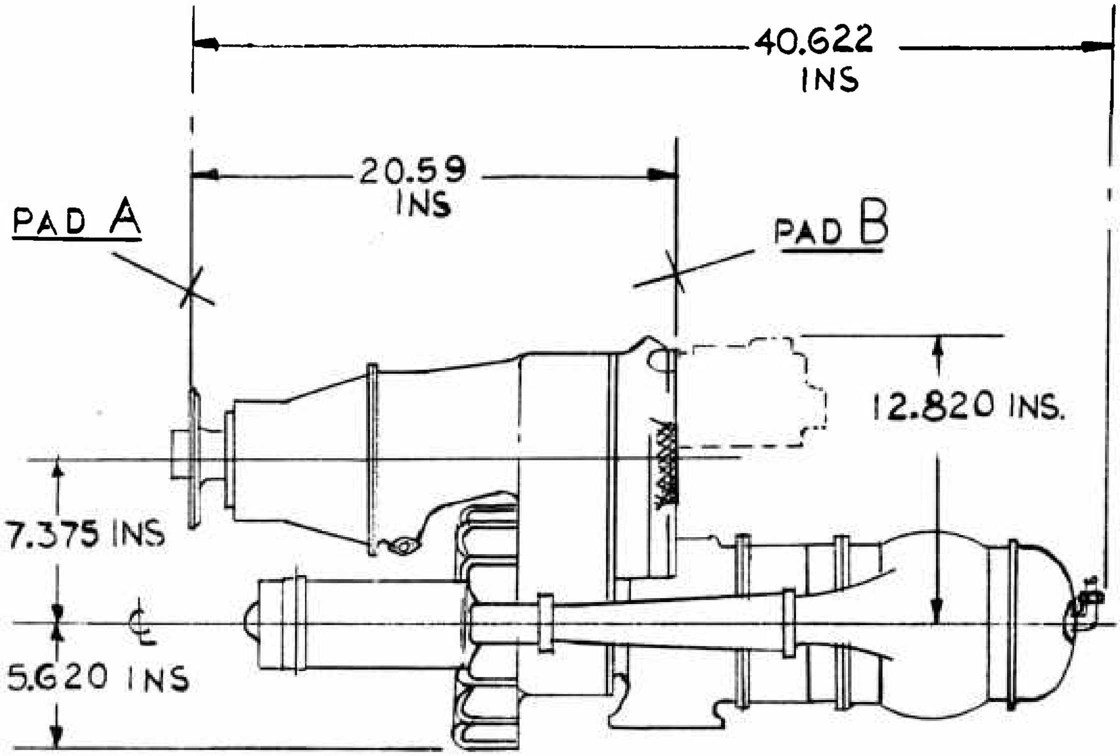
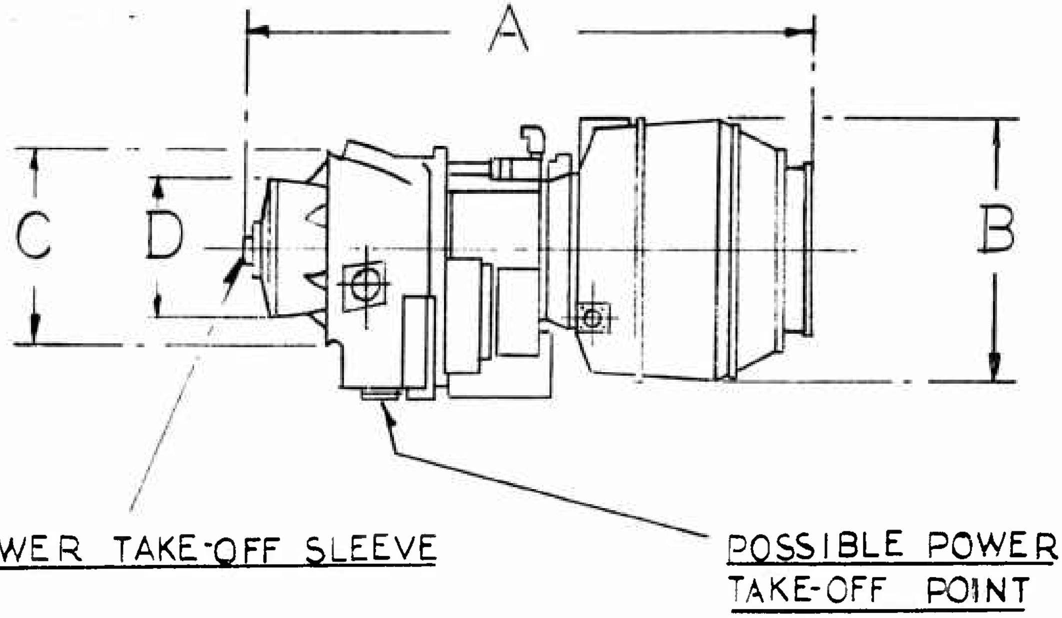


Figure 26. Differential Cam Transmission Housing Diameter vs Ratio



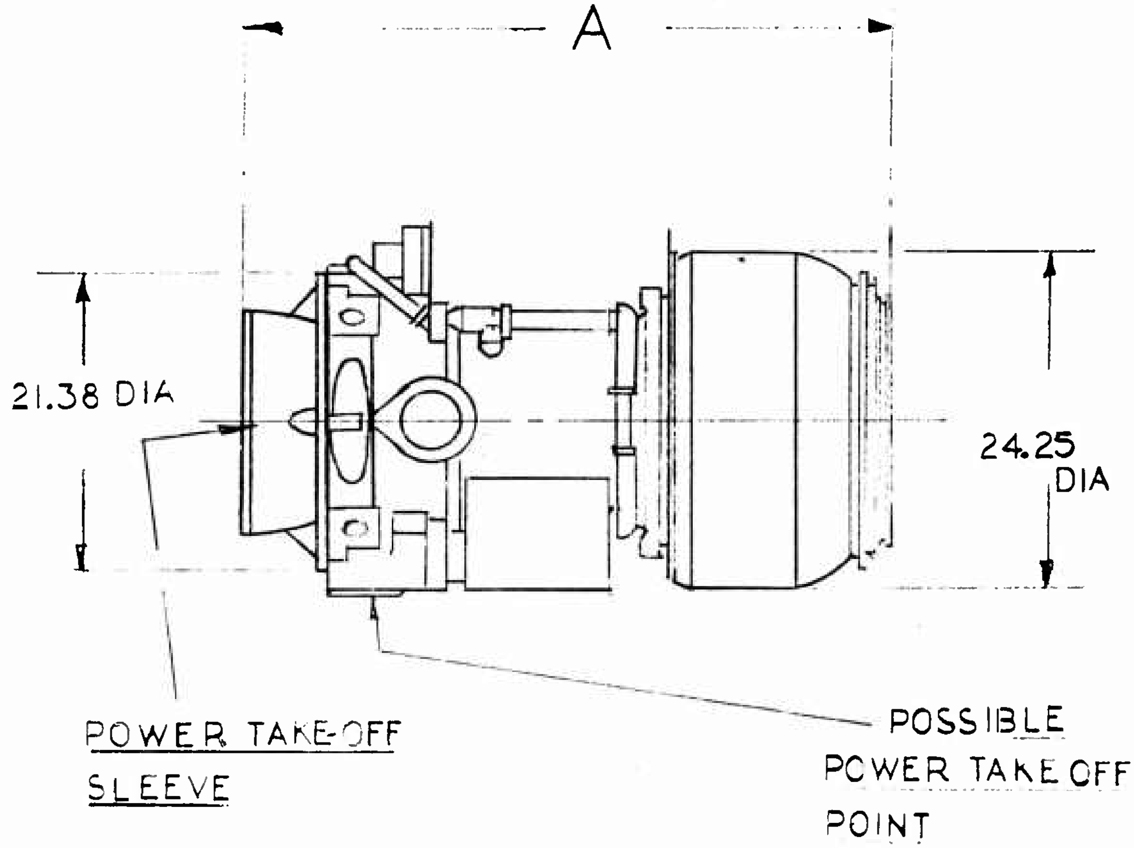
250 HORSEPOWER CAN BE EXTRACTED IN ANY COMBINATION OF A+B = 250 H.P. OR LESS.

Figure 27. Allison TG3 Turboshaft Engine



ENGINE	MAKER	A (INS)	B DIA (INS)	C DIA (INS)	D DIA (INS)
T53L1	LYCOMING	47.61	23.00	17.00	11.933
T53L5	LYCOMING	47.61	23.00	19.38	14.47

Figure 28. Lycoming T53 Turboshaft Engines



ENGINE	MAKER	A (INS)
T55 L3	LYCOMING	44.027
T55 L5	LYCOMING	44.040

Figure 29. Lycoming T55 Turboshaft Engines

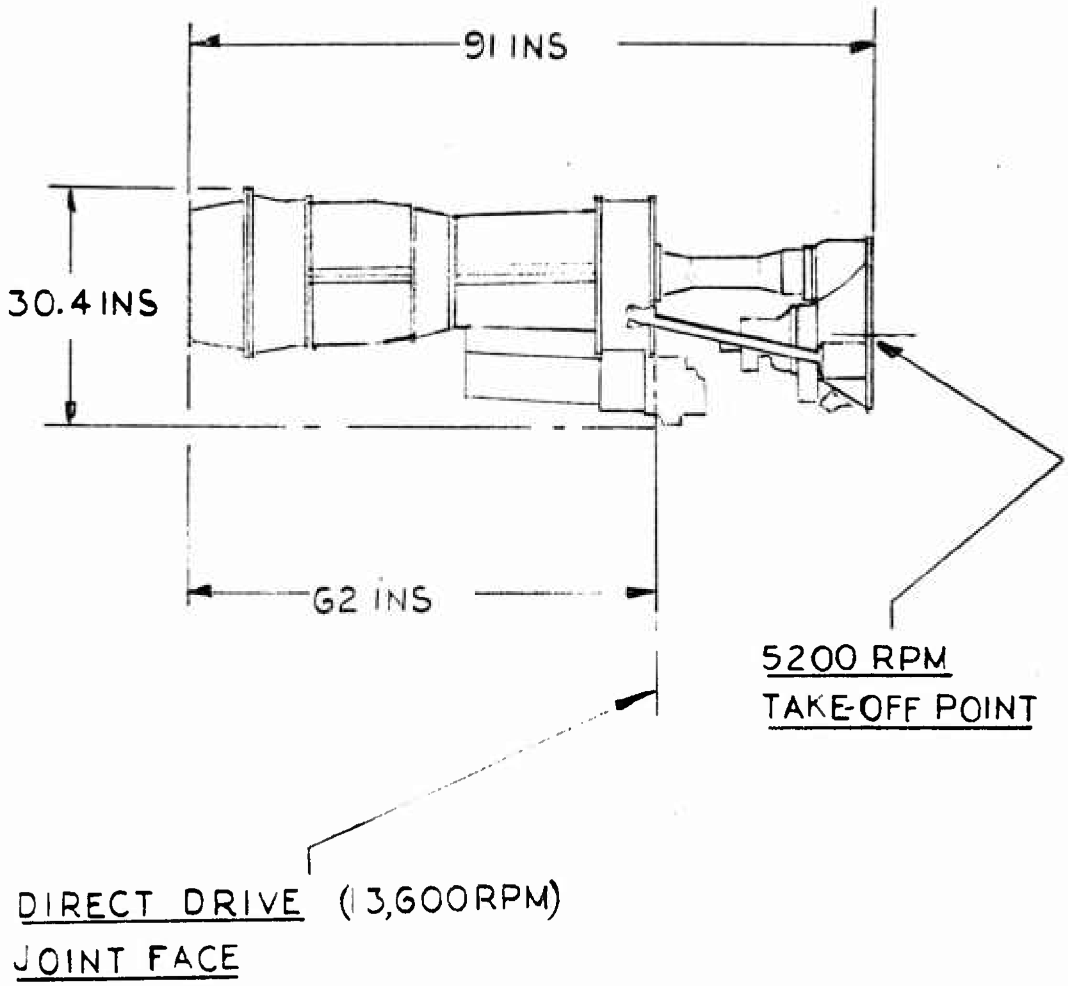


Figure 30. General Electric TG4-GE-2
Turboshaft & Direct Drive Engines

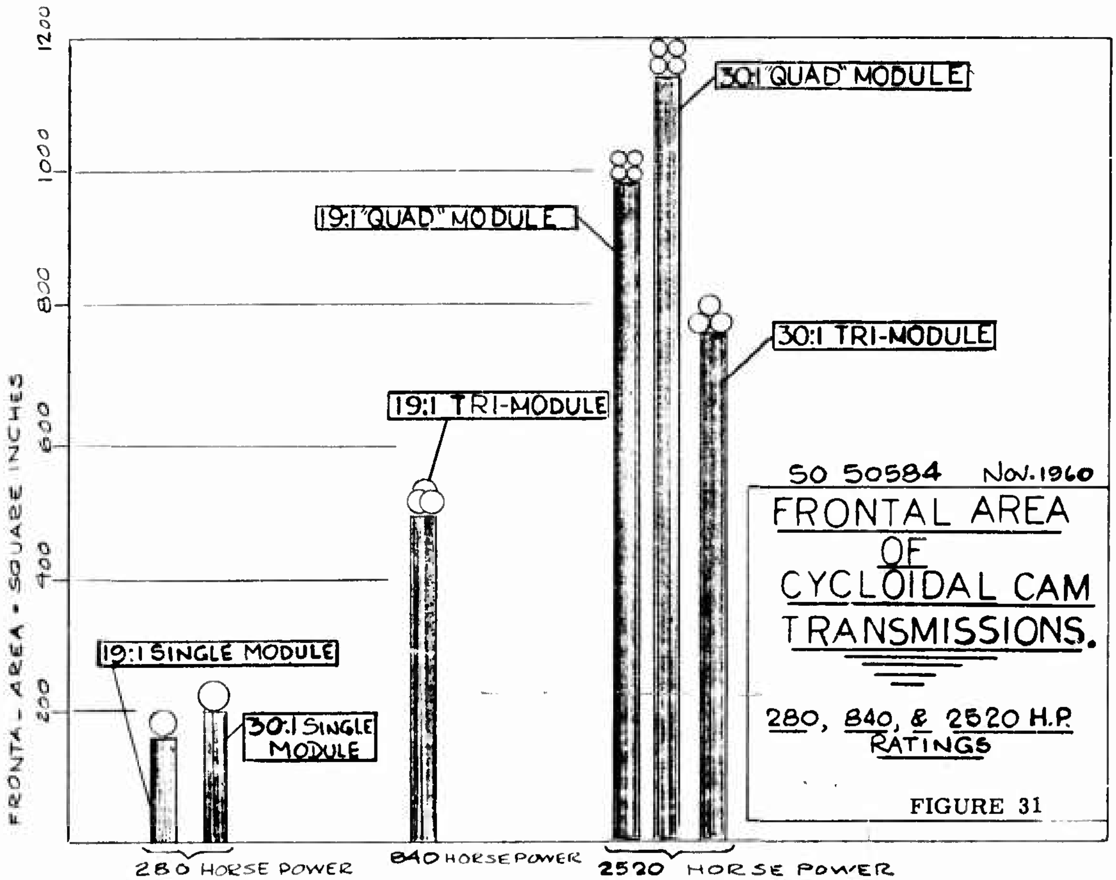

FRONTAL - AREA - SQUARE INCHES
1200
1000
800
600
400
200
30:1 "QUAD" MODULE
19:1 "QUAD" MODULE
30:1 TRI-MODULE
19:1 TRI-MODULE
19:1 SINGLE MODULE
30:1 SINGLE MODULE
SO 50584 Nov. 1960
FRONTAL AREA OF CYCLOIDAL CAM TRANSMISSIONS.
280, 840, & 2520 H.P. RATINGS
280 HORSE POWER
840 HORSEPOWER
2520 HORSE POWER

FIGURE 31

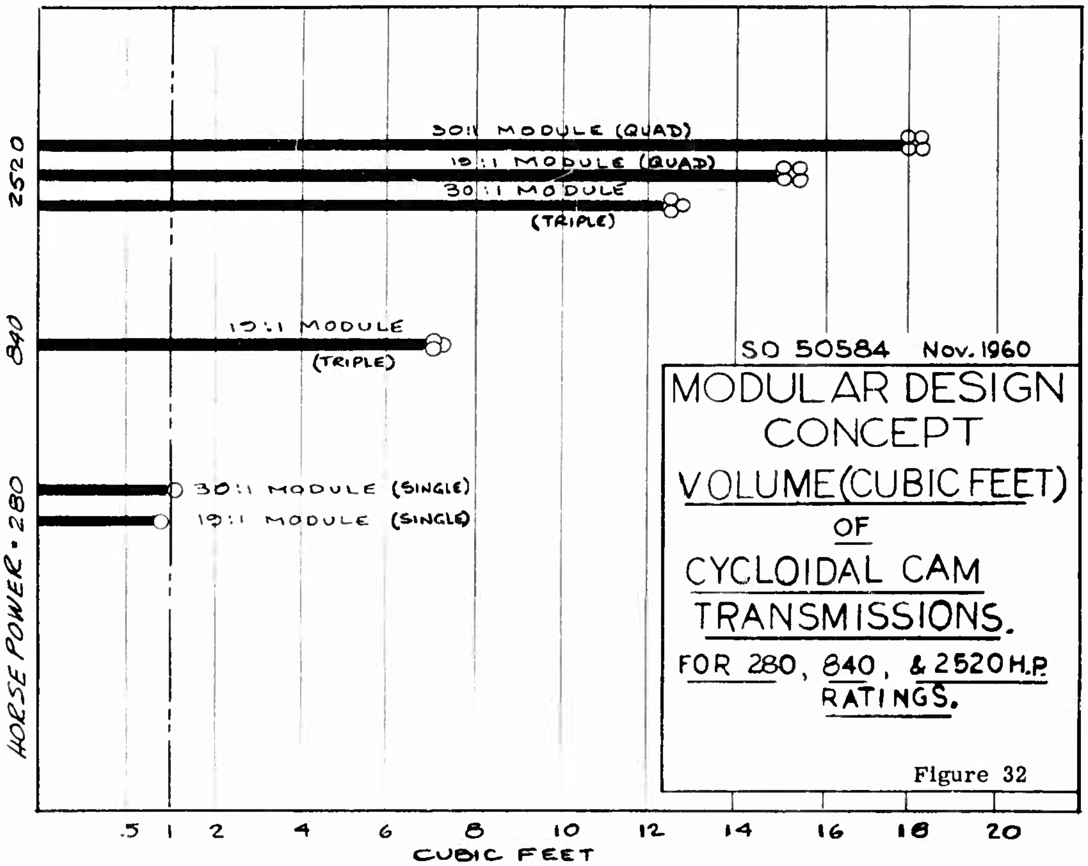

30:1 MODULE (QUAD)
15:1 MODULE (QUAD)
30:1 MODULE (TRIPLE)
15:1 MODULE (TRIPLE)
30:1 MODULE (SINGLE)
15:1 MODULE (SINGLE)
HORSE POWER - 280
840
2520
.5
1
2
4
6
8
10
12
14
16
18
20
CUBIC FEET
SO 50584 Nov. 1960
MODULAR DESIGN CONCEPT
VOLUME (CUBIC FEET)
OF
CYCLOIDAL CAM TRANSMISSIONS.
FOR 280, 840, & 2520 H.P. RATINGS.

Figure 32

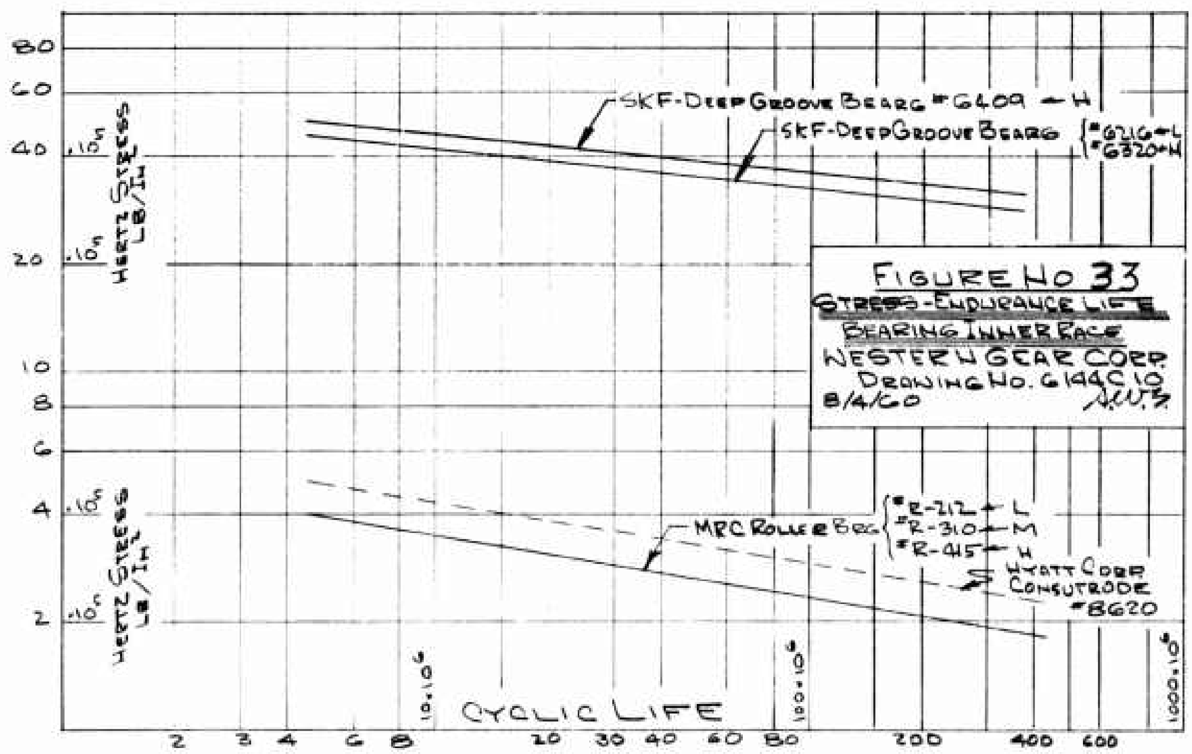
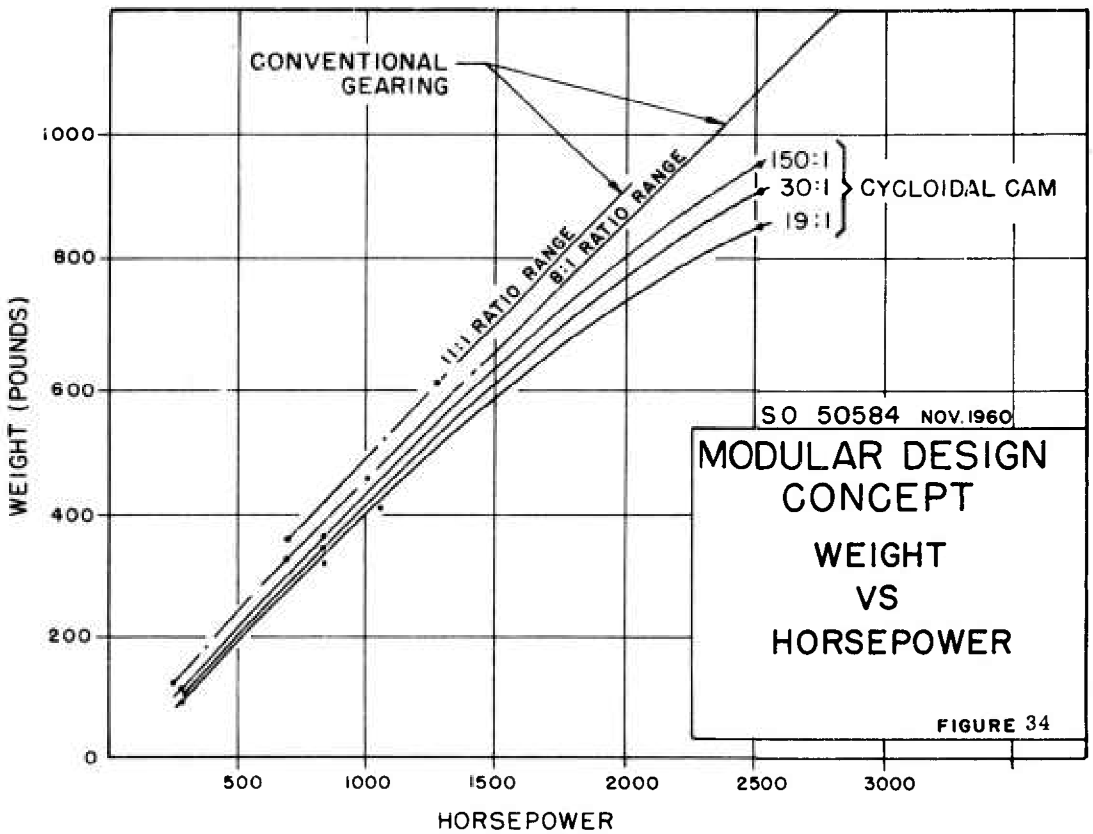
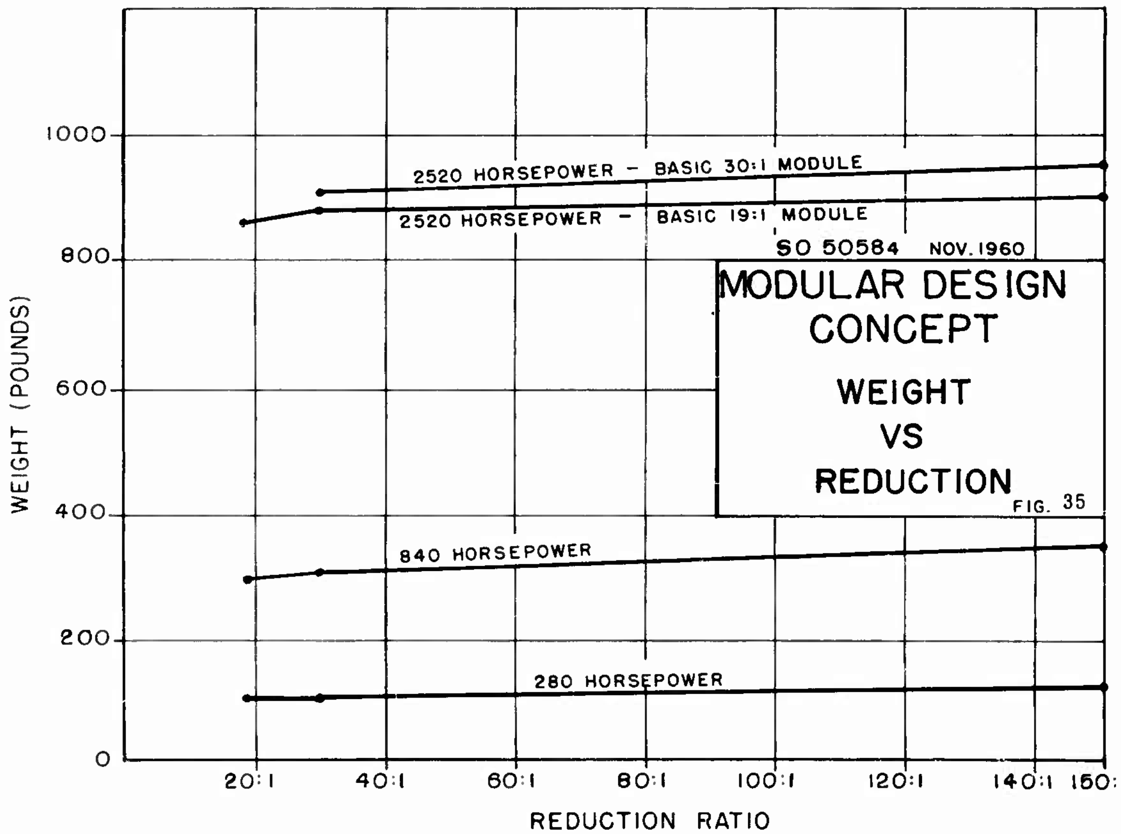
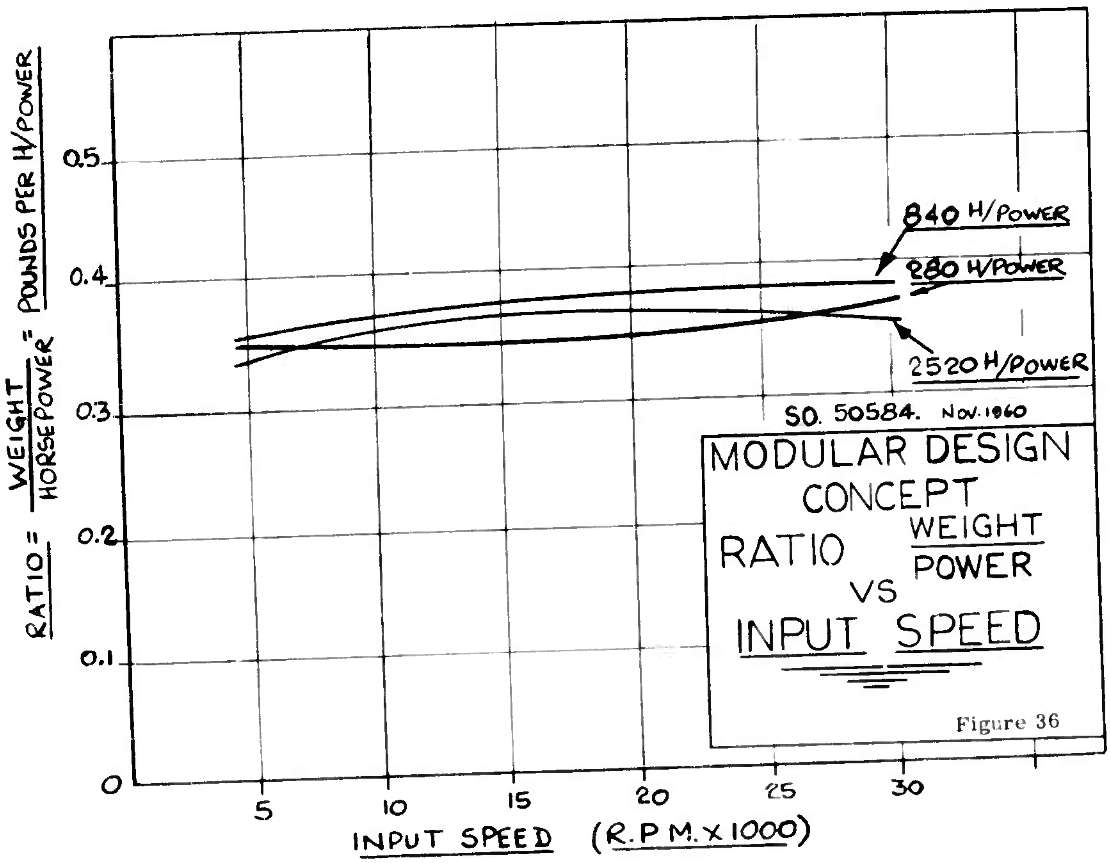


FIGURE NO 33
STRESS-ENDURANCE LIFE
BEARING INNER RACE
WESTERN GEAR CORP.
DRAWING NO. 6144C10
8/4/60
SKF-DEEP GROOVE BEARG #6409 — H
SKF-DEEP GROOVE BEARG #6216—L #6320—H
MRC ROLLER BRG #R-212 — L #R-310 — M #R-415 — H
HYATT CORP CONSUTRODE #8620
HERTZ STRESS LB/IN²
CYCLIC LIFE


CONVENTIONAL GEARING
11:1 RATIO RANGE
8:1 RATIO RANGE
150:1
30:1
19:1
CYCLOIDAL CAM
SO 50584 NOV. 1960
MODULAR DESIGN CONCEPT
WEIGHT VS HORSEPOWER
WEIGHT (POUNDS)
HORSEPOWER
0
200
400
600
800
1000
500
1000
1500
2000
2500
3000
FIGURE 34

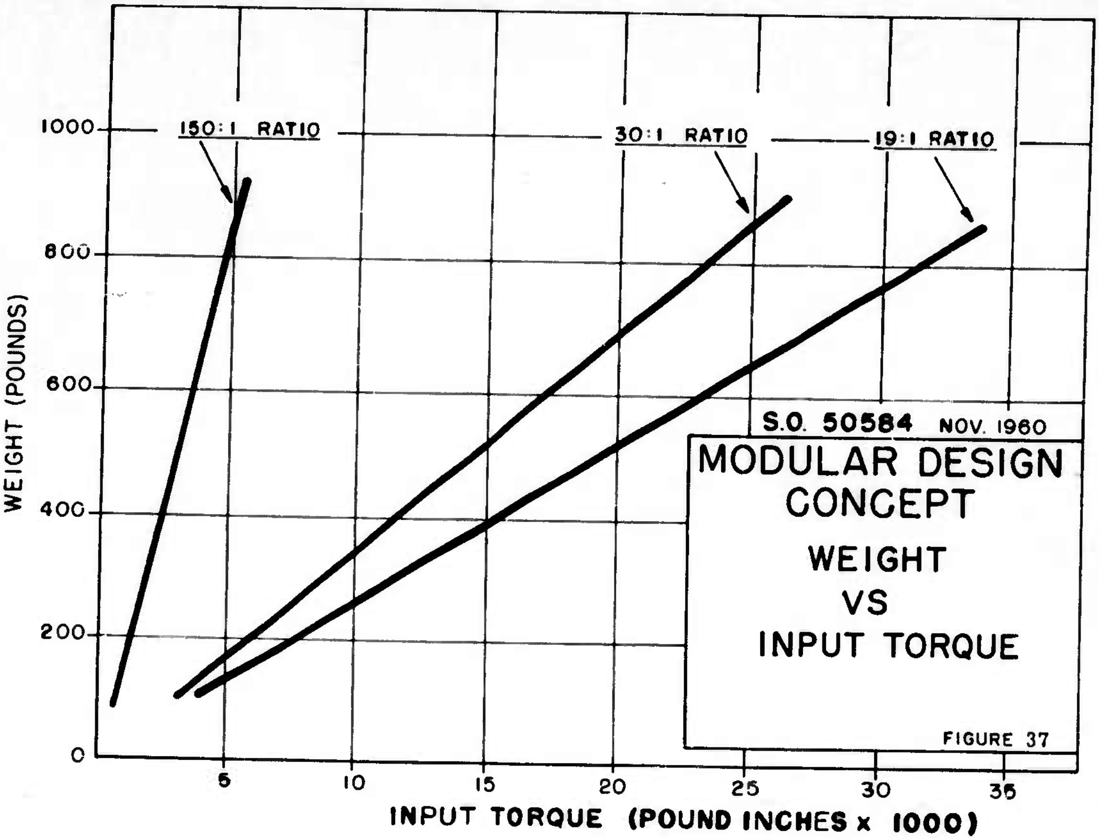


1000
800
600
400
200
0
WEIGHT (POUNDS)
2520 HORSEPOWER – BASIC 30:1 MODULE
2520 HORSEPOWER – BASIC 19:1 MODULE
SO 50584 NOV. 1960
MODULAR DESIGN CONCEPT
WEIGHT VS REDUCTION
FIG. 35
840 HORSEPOWER
280 HORSEPOWER
20:1
40:1
60:1
80:1
100:1
120:1
140:1
150:1
REDUCTION RATIO

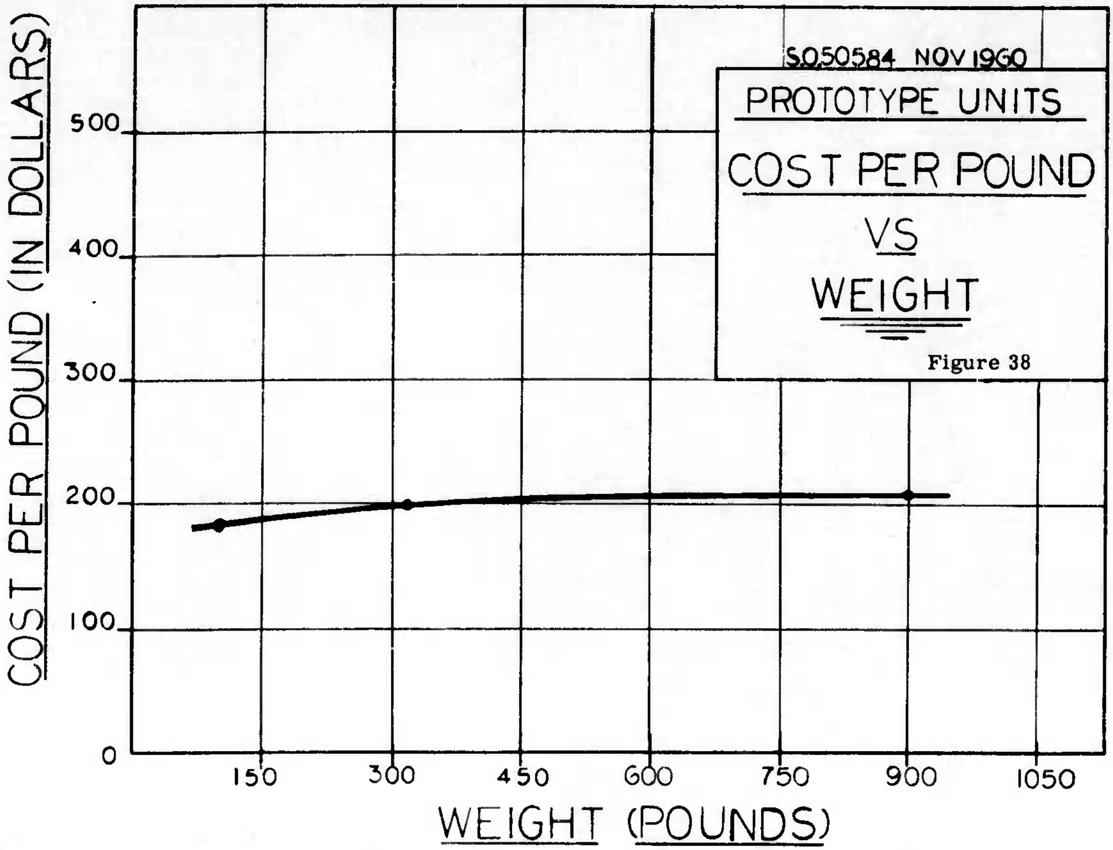


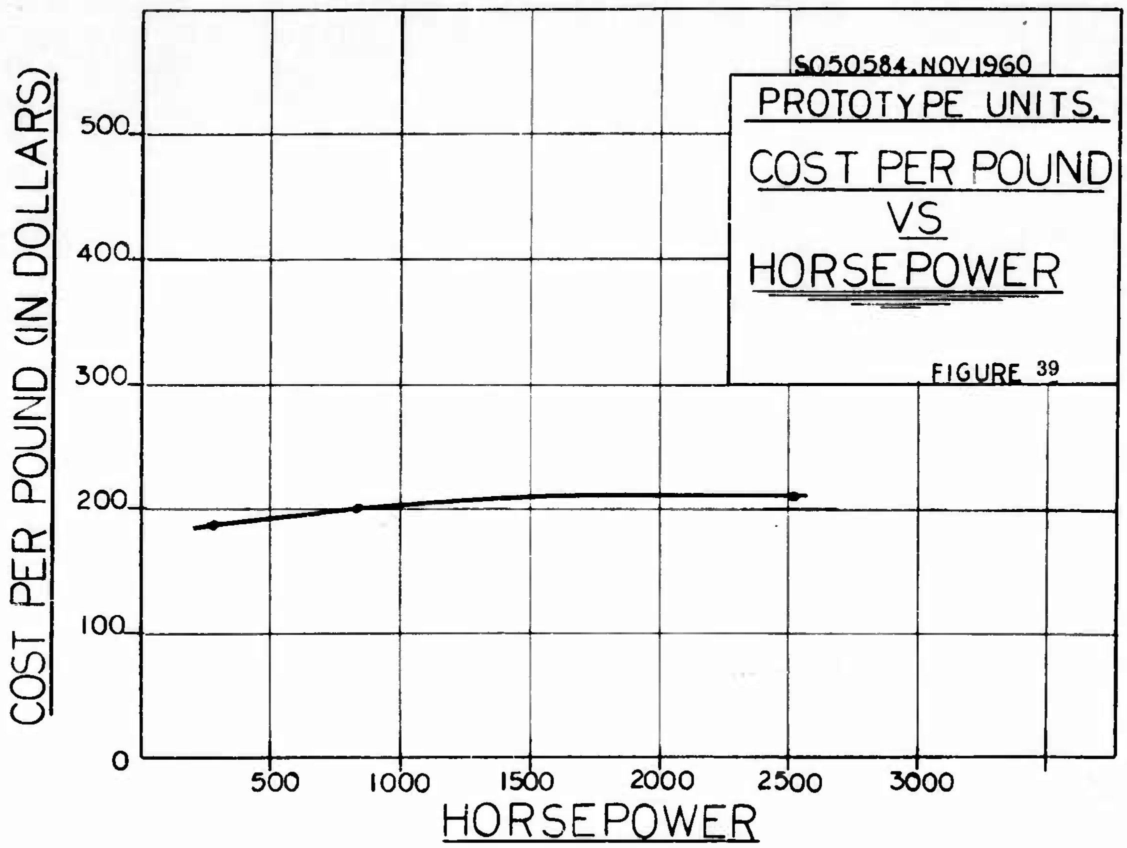
RATIO = WEIGHT / HORSEPOWER = POUNDS PER H/POWER
0.5
0.4
0.3
0.2
0.1
0
5
10
15
20
25
30
INPUT SPEED (R.P.M. x 1000)
840 H/POWER
280 H/POWER
2520 H/POWER
SO. 50584. Nov. 1960
MODULAR DESIGN CONCEPT
RATIO WEIGHT / POWER VS INPUT SPEED

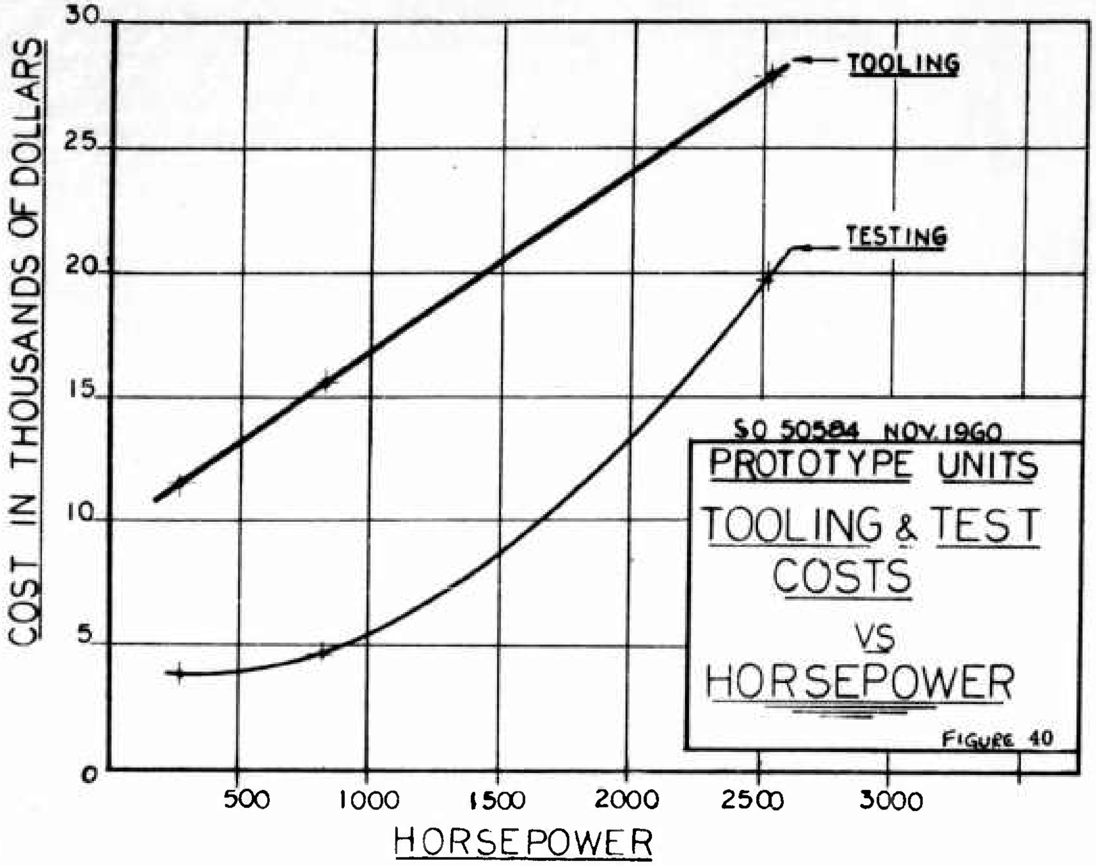
Figure 36



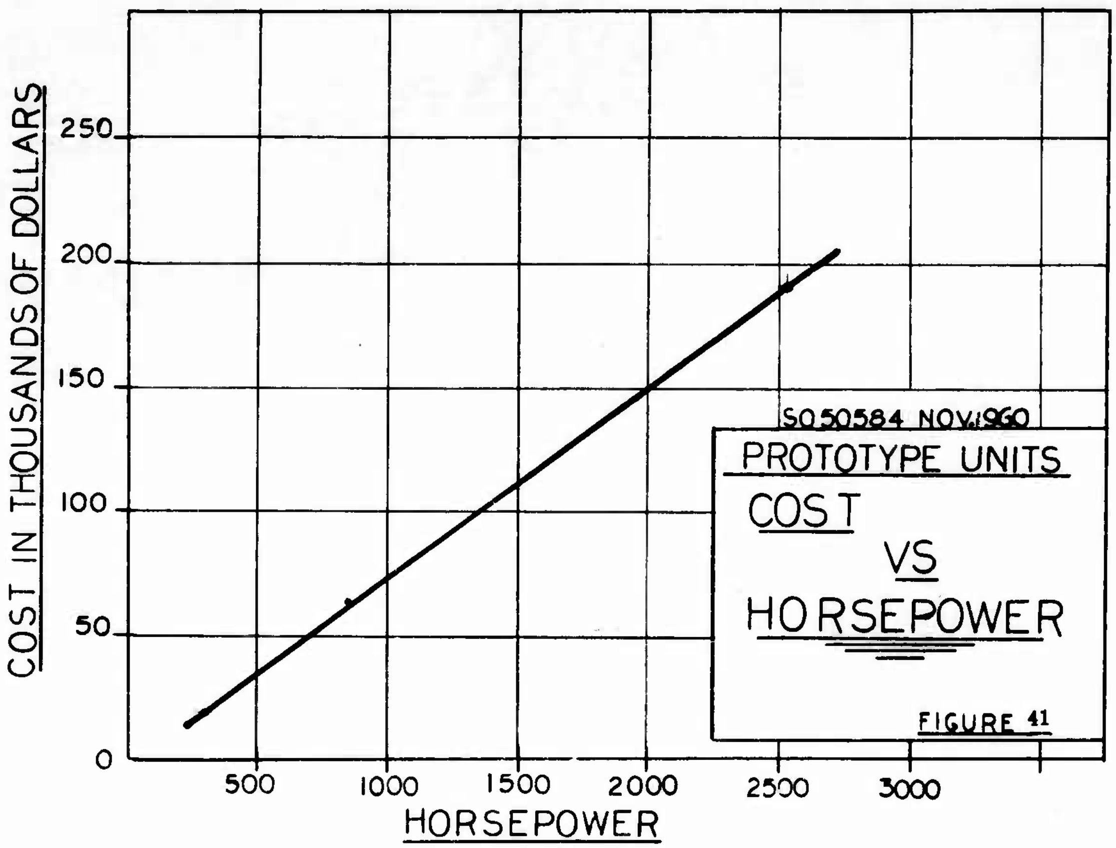
150:1 RATIO
30:1 RATIO
19:1 RATIO
1000
800
600
400
200
0
WEIGHT (POUNDS)
5
10
15
20
25
30
35
INPUT TORQUE (POUND INCHES x 1000)
S.O. 50584 NOV. 1960
MODULAR DESIGN CONCEPT
WEIGHT VS INPUT TORQUE
FIGURE 37


S050584 NOV 1960
PROTOTYPE UNITS
COST PER POUND
VS
WEIGHT
COST PER POUND (IN DOLLARS)
500
400
300
200
100
0
150
300
450
600
750
900
1050
WEIGHT (POUNDS)
Figure 38


S050584.NOV1960
PROTOTYPE UNITS.
COST PER POUND
VS
HORSEPOWER
FIGURE 39
COST PER POUND (IN DOLLARS)
0
100
200
300
400
500
500
1000
1500
2000
2500
3000
HORSEPOWER



COST IN THOUSANDS OF DOLLARS
30
25
20
15
10
5
0
TOOLING
TESTING
SO 50584 NOV. 1960
PROTOTYPE UNITS
TOOLING & TEST
COSTS
VS
HORSEPOWER
FIGURE 40
500
1000
1500
2000
2500
3000
HORSEPOWER



250
200
150
100
50
0
COST IN THOUSANDS OF DOLLARS
500
1000
1500
2000
2500
3000
HORSEPOWER
SO50584 NOV.1960
PROTOTYPE UNITS
COST
VS
HORSEPOWER
FIGURE 41

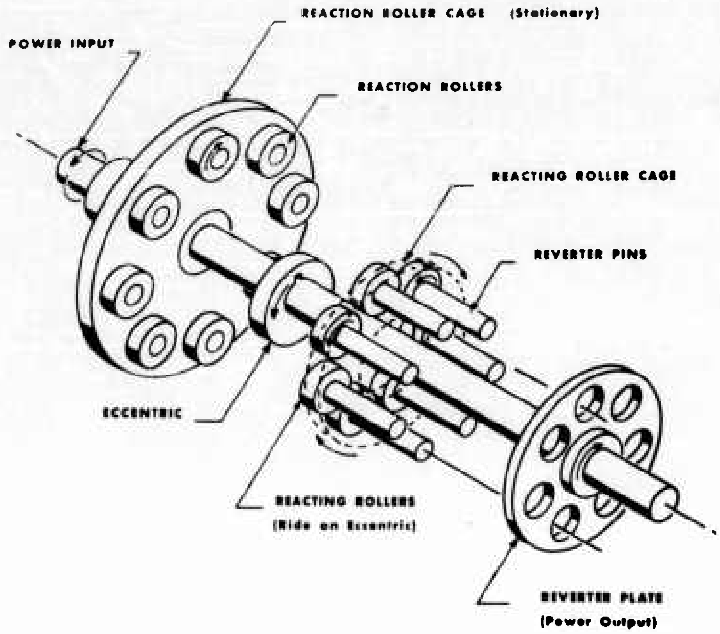


Figure 42. Cycloidal Roller Transmission

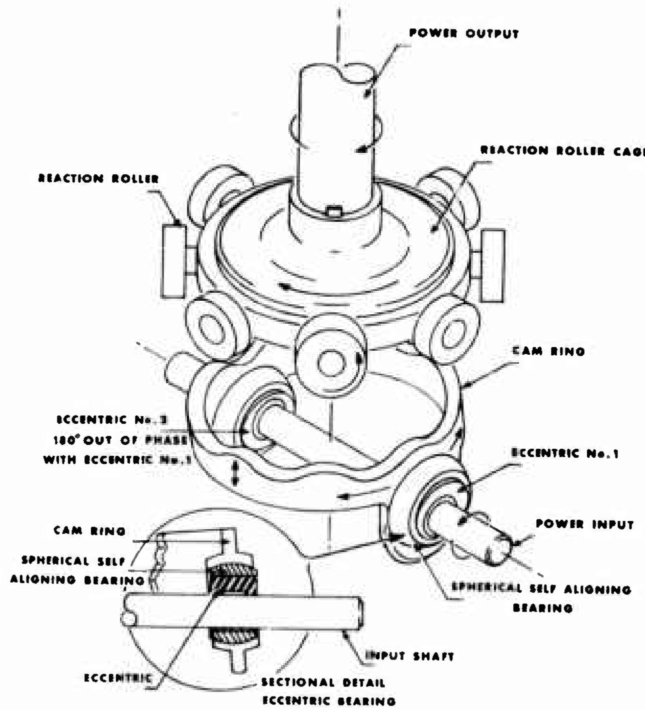


Figure 43. Right Angle Cycloidal Roller Transmission

TABLES

TABLE 1
CAM AND FOLLOWER
KINEMATIC DATA AND GEOMETRICAL RESULTS

Motion of follower	Zero velocity	Constant velocity	Constant acceleration	Simple harmonic motion	
				One cycle	Two cycles
Co-ordinate displacements of center of curvature in direction of:					
Tangent	Zero	Constant	Increasing	Harmonic displacement	Harmonic displacement
Normal	Zero	Constant	Constant		
Position of normal	Fixed, at center	Fixed, offset	Changing at constant velocity	Harmonic displacement	Harmonic displacement
Center of curvature relative to frame	Stationary, at center	Stationary, offset	Moving on offset line	Displaced around circle	Displaced around ellipse
Evolute of cam profile	Point	Circle	Involute of circle	Point	Hypocycloid of four cusps
Cam profile	Circle	Involute of circle	Involute of involute of circle	Circle	Involute of hypocycloid
Instant center	At center of rotation	Fixed	Moving along line of centers at constant velocity	Harmonic displacement	Harmonic displacement
Centrode of follower	Straight line, on center	Straight line, offset	Parabola	Circle	Ellipse
Centrode of cam	Point, at center	Circle	Spiral of Archimedes	Circle	Four-looped hypocycloid

NOTE: Data from Machine Design February 1947

TABLE NO. 2[illegible]
SINGLE STAGE CYCLOID CAM TRANSMISSION

Horsepower	Ratio	Eccentric	Cam Plate Diameter
280	29:1	.200	15 inches
280	59:1	.187	28 inches
280	89:1	.160	34 inches
280	149:1	.109	43 inches
840	29:1	.250	22 inches
840	59:1	.200	32 inches
840	89:1	.170	41 inches
840	149:1	.125	50 inches
1680	29:1	.297	25 inches
1680	59:1	.247	40 inches
1680	89:1	.180	43 inches
1680	149:1	.135	54 inches
2520	29:1	.375	31 inches
2520	59:1	.297	47 inches
2520	89:1	.218	54 inches
2520	149:1	.157	62 inches

TABLE NO. 3
SINGLE STAGE CYCLOID CAM TRANSMISSION

HORSEPOWER	RATIO	WEIGHT
280	29:1	100
280	59:1	190
280	89:1	265
280	149:1	286
840	29:1	350
840	59:1	520
840	89:1	667
840	149:1	815
1680	29:1	800
1680	59:1	1300
1680	89:1	1400
1680	149:1	1760
2520	29:1	1400
2520	59:1	2120
2520	89:1	2440
2520	149:1	2800

TABLE NO. 4
SINGLE STAGE ROLLER CAM TRANSMISSION
WEIGHT & HOUSING DIAMETER vs HORSEPOWER

HORSEPOWER	RATIO	HOUSING DIAMETER	WEIGHT
280	29:1	17 inches	99 pounds
280	59:1	30 "	264 "
280	89:1	36 "	442 "
280	149:1	45 "	550 "
840	29:1	24 inches	422 pounds
840	59:1	34 "	528 "
840	89:1	43 "	700 "
840	149:1	52 "	---- -
1680	29:1	29 inches	824 pounds
1680	59:1	43 "	886 "
1680	89:1	46 "	935 "
1680	149:1	56 "	---- -
2520	29:1	35 inches	1058 pounds
2520	59:1	50 "	1151 "
2520	89:1	56 "	1221 "
2520	149:1	65 "	1438 "

TABLE NO. 5
DIFFERENTIAL CAM DATA

Horsepower	Reduction Ratio	Eccentricity	Housing Diameter Stage Phase Input	Output	Configuration Cam Lobes N_1 X N_2	Weight Pounds
280	29:1	.200	19"	16"	15 X 31	185
	59:1	.166	20"	17"	21 X 33	207
	89:1	.152	20"	18"	23 X 31	208
	149:1	.125	20"	19"	27 X 33	225
840	29:1	.312	21"	31"	15 X 31	864
	59:1	.250	18"	26"	21 X 33	655
	89:1	.218	17"	24"	23 X 31	610
	149:1	.187	18"	25"	27 X 33	630
1680	29:1	.312	38"	48"	21 X 73	2700
	59:1	.281	36"	44"	25 X 45	1550
	89:1	.204	34"	43"	29 X 43	1480
	149:1	.168	33"	40"	37 X 49	1500
2520	29:1	.312	47"	51"	21 X 73	3620
	59:1	.297	43"	41"	25 X 45	2130
	89:1	.190	33"	45"	57 X 61	2105
	149:1	.150	37"	41"	43 X 61	2120

TABLE 6
POWER SOURCE ANALYSIS

Engine	Maker	Rotor Type	Application	S.H.P. (Normal Rating)	Residual Thrust (lb)	Rotor Speed (rpm)	Output Ratio	Output Speed (rpm)	Weight (lb)
T.53L3 *	Lycoming	Free Turbine	Turbo-Prop	825	100	21,500	0.0802:1 (12.46:1)	1720	530
T.53L1 *	Lycoming	Free Turbine	Helicopter	770	96	21,500	0.310:1 (3.22:1)	6680	480
T.55L1 *	Lycoming	Free Turbine	Turbo-Prop	1325	168	13,500	0.097:1 (10.24:1)	1320	695
T.55L3 *	Lycoming	Free Turbine	Helicopter	1650	194	14,500	0.466:1 (2.1475:1	6720	600
T.63 *	Allison	Free Turbine	Turbo Shaft	250	-	27,000	0.2222:1 (4.5/1)	6000	90
T.64 Δ G.E.4	General Electric	Free Turbine	Turbo-Prop	2230	-	13,600	0.085:1 (11.8:1)	1150	1079
T.64 Δ G.E.2	General Electric	Free Turbine	Turbo-Shaft	2235	-	13,600	0.382:1 (2.62:1)	5200	854

* P.H. Wilkinson "Aircraft Engines of the World - 1958/59." Paul H. Wilkinson Washington, D.C.
Δ "The General Electric T.64" - Brochure May 1959. General Electric, Lynn, Mass.

Table 6
POWER SOURCE ANALYSIS

TABLE 7
POWER SOURCE ANALYSIS

Engine	Maker	Version	Shaft Horsepower (Normal)	Engine Rotor Speed (R. P. M.)	Normal Output Speed (R. P. M.)	Approx. Reduction Ratio Required for 200 R. P. M. @ Helicopter Rotor (Direct Drive)	Approx. Reduction Ratio Required for 200 R. P. M. @ Helicopter Rotor (Normal Output)
T. 63*	Allison	Turboshaft	250	27,000	6000	135/1	30/1
T. 53. L1*	Lycoming	Helicopter	770	21,500	6610	107/1	33/1
T. 53L5*	Lycoming	Helicopter	825	21,500	6610	107/1	33/1
T. 55L3*	Lycoming	Helicopter	1650	14,500	6720	72/1	33/1
T. 55L5*	Lycoming	High Speed	1735	13,870	-	68/1	-
T. 64** G. E. 2	General Electric	Helicopter	2230	13,600	5200	68/1	26/1

* P. H. Wilkinson "Aircraft Engines of the World", 1958/59 Paul H. Wilkinson, Washington, D. C.

** "The General Electric T. 64" - Brochure, May 1959 General Electric, Lynn, Massachusetts

TABLE 8
WEIGHT ANALYSIS
280 HP MODULE

280 HORSEPOWER MODULE

29:1 Cycloid Reduction

Major Component	Weight Pounds
Input Shaft	3.608
Output Flange Shaft	3.023
Sub-Shaft	1.2105
Fixed Pin Rollers and Bearings	7.680
Fixed Pins	4.50
Spacers	.3654
Reverter Pin Eccentrics, small	2.10
Reverter Pin Eccentrics, large	2.58
Bearing Outer Races, small	5.76
Bearing Outer Races, large	5.04
Reverter Needle Bearings, large	2.94
Reverter Needle Bearings, small	.9112
Reverter Pins	2.1654
Cam Rings	6.704
Center Cam	6.704
Bearings - Cam (outer)	1.064
Bearings - Cam (inner)	5.024
Input Bearing	1.340
Output Bearing	.960
2 Spinners @ .39 pounds	.780
2 (Inner Shaft)	.270
Cam Webs (outer)	2.380
Center Cam	4.760
Output Plates	3.326
Housing	22.000
TOTAL WEIGHT	98.783

19:1 Cycloid Reduction

Major Component	Weight Pounds
Center Shaft	5.429
Lock and Thrust Washers	.063
Nut	.114
Gear Input	1.584
Reverter Pin Output Spider (Idler)	3.946
Cam - Input Side	12.648
Cam - Center	22.718
Cam - Output Side	12.648
Reverter Pin Output Spider	3.146
Gear Output	2.543
Spacers on Center Shaft	2.002
Output Pins (12)	3.268
Reverter Pins (20)	11.040
	81.149
Housing	17.000
TOTAL WEIGHT	98.149

840 HORSEPOWER TRANSMISSION

30:1 Cycloid Reduction

Major Component	Weight Pounds
19:1 Cycloid Reduction (3)	243.0
Housing	53.8
Shafting and Gears	11.0
TOTAL WEIGHT	307.8

TABLE 9
TRANSMISSION DRY WEIGHT ANALYSIS

Transmissions Helicopter Model	Dry Weight (Less Accessories) LB	Applied B. H. P.	Ratio LB Per HP $= \frac{WT}{HP}$	Reduction Ratio
H. 13	* 92.4	200	.46	
H. 19	* 325	700	.464	
H. 21(Fwd)	* 361	712	.506	Range
H. 21(Aft)	* 361	712	.506	
H. 23	* 123	250	.492	from
H. 34	* 610	1275	.48	
H. 37	*1600	4200	.38	8 to 11:1
HU. 1A	* 332	700	.43	
HU. 1B	* 364	1100	.33	
HU. 1D	* 384	1100	.349	
Cycloidal Cam Prelim. Designs				
6144R43	** 99	280	.353	19:1
***	** 300	840	.358	19:1
6144D48	** 857	2520	.340	19:1
6144R45	** 98	280	.350	30:1
6144D51	** 307	840	.366	30:1
6144D48	** 911	2520	.361	30:1
***	** 880	2520	.350	30:1
***	** 105	280	.375	150:1
***	** 317	840	.377	150:1
6144D48	** 954	2520	.379	150:1
6144D49	** 900	2520	.358	150:1

* Actual weights
** Estimated weights
*** Postulated

TABLE 10
HELICOPTER DRY WEIGHT ANALYSIS *

Helicopter Model	Transmission Dry Weight	Max. Power Applied (BHP)	Weight of Accessories	Weight of Lubricants
H-13	94.5	200	Unknown	Engine Oil Used
H-19	328.0	700	82.7	20.6
H-21(FWD)	381.0	712	21.0	15.9
H-21(MID)	190.0	1425	20.0	9.4
H-21(AFT)	359.0	712	0	15.9
H-23	122.7	280	Unknown	Engine Oil Used
H-34	610.2	1295	66.3	22.2
H-37(MAIN)	1600.0	4200	35.0	50.0
H-37(MID)	79.0	300	0	10.0
H-37(TAIL)	160.0	300	0	6.0
HU-1A	346.0	770	58.2	14.1
HU-1B	364.4	960	58.0	18.1
HU-1D	384.4	1100	58.0	18.1

* Information furnished by J. Wallace McDonald Deputy for Aviation, Directorate of Engineering, U.S. Army Transportation Material Command, St. Louis, Missouri (18 August, 1960) & TRECOM, Ft. Eustis, Virginia.

TABLE 11
MATERIAL RECOMMENDATIONS*

ITEM	MATERIAL	TREATMENT	HARDNESS (R_c)	CONDITION
Input Gear	4620	Carburize and harden	60-63	Case depth .018" to .024"
Jack Shaft	4340	Through harden	38-43	
Crank Shaft	4620	Carburize and harden	60-63 core 32(min)	Case depth .035" to .045"
Cam Plate	4620(Vacuum melted)	Carburize and harden	60-63	Case depth .035" to .045"
Cam Plate Adapter	7075T6	Stress Relieve	-----	Aluminum forging
Reaction Pins	9310	Carburize and harden	60-63	Case depth .020" to .025"
Reverter Pins	9310	Carburize and harden	60-63	Case depth .020" to .025"
Main Crank Bearing	52100(Vacuum Melted)	Through Harden	61-64	
Bearing Races	52100(Vacuum Melted)	Through Harden	61-64	
Output Gear & Spline	9310	Carburize and harden	60-63 core 30(min.)	Gear case .040" to .050" Spline case .020" to .025"
Housing	356-T6	Stress Relieve	-----	Aluminum casting

* Based on Drawing 6144R43

TABLE 12
PATENT EVALUATION

Patent Number	FIGURE OF MERIT wt	mfr	cost	dev	eff	torque	noise	score
1,828,795	2	2	3	3	3	4	3	20
2,666,345	2	2	3	3	3	4	3	20
2,520,282	5	3	3	3	4	5	3	26
Re. 17,811	5	3	3	3	5	4	5	28
1,692,160	4	3	3	3	4	4	4	25
2,475,504	5	5	4	4	4	5	3	30
1,942,795	3	4	3	4	3	4	3	24
1,867,492	4	4	3	3	5	4	5	28
1,910,777	1	1	1	1	3	4	3	14
2,239,839	1	4	4	4	3	5	3	24
2,874,594	3	2	1	2	4	4	3	19
1,449,352	2	3	2	3	4	4	4	22
1,116,970	1	4	4	4	3	5	3	24
1,773,568	3	3	3	4	4	3	4	24
Re. 24,288	4	4	4	3	4	4	4	28
1,870,875	4	3	3	3	5	4	5	27
1,694,031	5	3	3	3	5	4	5	28
1,767,866	2	3	3	4	3	5	3	23
2,529,997	2	2	3	3	2	3	4	19
1,738,662	4	2	2	2	2	5	3	20
2,677,288	3	3	3	3	4	5	3	24
2,508,121	3	4	4	4	4	3	3	25
498,552	1	2	3	3	1	3	3	16
1,641,766	5	4	5	5	5	5	4	33

Key for Table 1 Explaining "Figure of Merit".

Areas of Evaluation	Rating System
1. Weight (wt)	5. Highest
2. Ease of Manufacture (mfr)	4.
3. Cost	3.
4. Improvements (dev)	2.
5. Efficiency (eff)	1. Least
6. Torque Limitations (torque)	
7. Noise (Noise)	

TABLE 13
BALL BEARING DATA

Bearing Number	P max (lbs)	Ball (dia)	F	$\left[\frac{P\ max}{d^2F}\right]^{1/3}$	H max* (psi)
6201	250	.070	9.2	7.33	434,000
6204	646	.097	10.5	8.56	509,000
6206	984	.145	11.1	8.44	501,500
6209	1442	.250	11.3	7.96	472,000
6212	2075	.390	11.5	7.55	448,000
6216	2860	.560	11.8	7.55	448,000
6220	4730	1.000	11.6	7.40	439,000
6224-X	6470	1.411	11.6	7.70	458,000
6234-X	7370	1.880	12.4	6.85	406,000
6240-X	9500	2.640	12.4	5.62	333,500
6302	646	.097	9.7	7.82	466,000
6305	1250	.191	10.1	8.65	514,000
6308	2030	.350	10.6	8.16	485,000
6311	3580	.660	10.7	7.95	472,000
6314	5050	1.000	10.6	7.80	464,000
6317	6680	1.305	10.7	7.80	464,000
6320	9150	2.070	10.6	7.45	442,000
6322	10920	2.640	10.5	7.35	436,000
6328-X	13800	3.500	10.9	7.13	423,000
6334-X	17050	4.780	11.1	6.85	417,000
6340-X	18700	5.625	11.4	6.62	394,000
6352-X	22100	7.530	11.8	6.20	368,000
6403	1670	.250	8.9	9.07	540,000
6406	2460	.429	9.7	8.40	500,000
6409	4460	.820	9.7	8.08	480,000
6412	6400	1.265	9.8	8.02	477,000
6415	9400	2.060	9.7	7.77	461,000
6418	10920	2.210	9.9	7.93	470,000

$$* H = 10^4 \times \left[210 \times \frac{P\ max}{d^2F}\right]^{1/3} = 10^4 \times 5.94 \times \left[\frac{P\ max}{d^2F}\right]^{1/3}$$

TABLE 14
BEARING DATA FOR SKF 6216, 6320 AND 6409

RPM	Load Ball	d/P.D.	H max (psi)	Stress Cycles
200 (6216)	656	.174	448,000	6×10^6
500	484	.174	408,000	15×10^6
1500	341	.174	362,000	45×10^6
5000	220	.174	313,000	150×10^6
200 (6320)	2090	.232	442,000	6×10^6
500	1540	.232	400,000	15×10^6
1500	1070	.232	354,000	45×10^6
3000	854	.232	329,000	90×10^6
200 (6409)	1020	.279	480,000	6×10^6
500	755	.279	442,000	15×10^6
1500	520	.279	391,000	45×10^6
5000	346	.279	342,000	150×10^6

TABLE 15
BEARING DATA FOR MRC R212, R-310 AND R-415

RPM	Load Roll	r/R	H max (psi)	Stress Cycles
200 (R-212)	688	.198	375,000	6×10^6
500	515	.198	308,000	15×10^6
1000	406	.198	275,000	30×10^6
5000	250	.198	216,000	150×10^6
200 (R-310)	835	.231	394,000	6×10^6
500	616	.231	316,000	15×10^6
1000	480	.231	284,000	30×10^6
3000	350	.231	242,000	90×10^6
200 (R-415)	3185	.293	388,000	6×10^6
500	2365	.293	319,000	15×10^6
1000	1878	.293	290,000	30×10^6
5000	1210	.293	226,000	150×10^6

TABLE 16
ROLLER BEARING DATA

Bearing Number	r/R	Hertz Max PSI
R-204	.265	468000
R-207	.230	392000
R-212	.198	375000
R-218	.168	352000
R-222	.182	348000
R-230	.179	367000
R-302	.327	448000
R-305	.260	415000
R-310	.231	394000
R-315	.246	365000
R-318	.237	381000
R-324	.245	374000
R-412	.280	398000
R-415	.293	388000
R-418	.296	398000
R-421	.282	386000

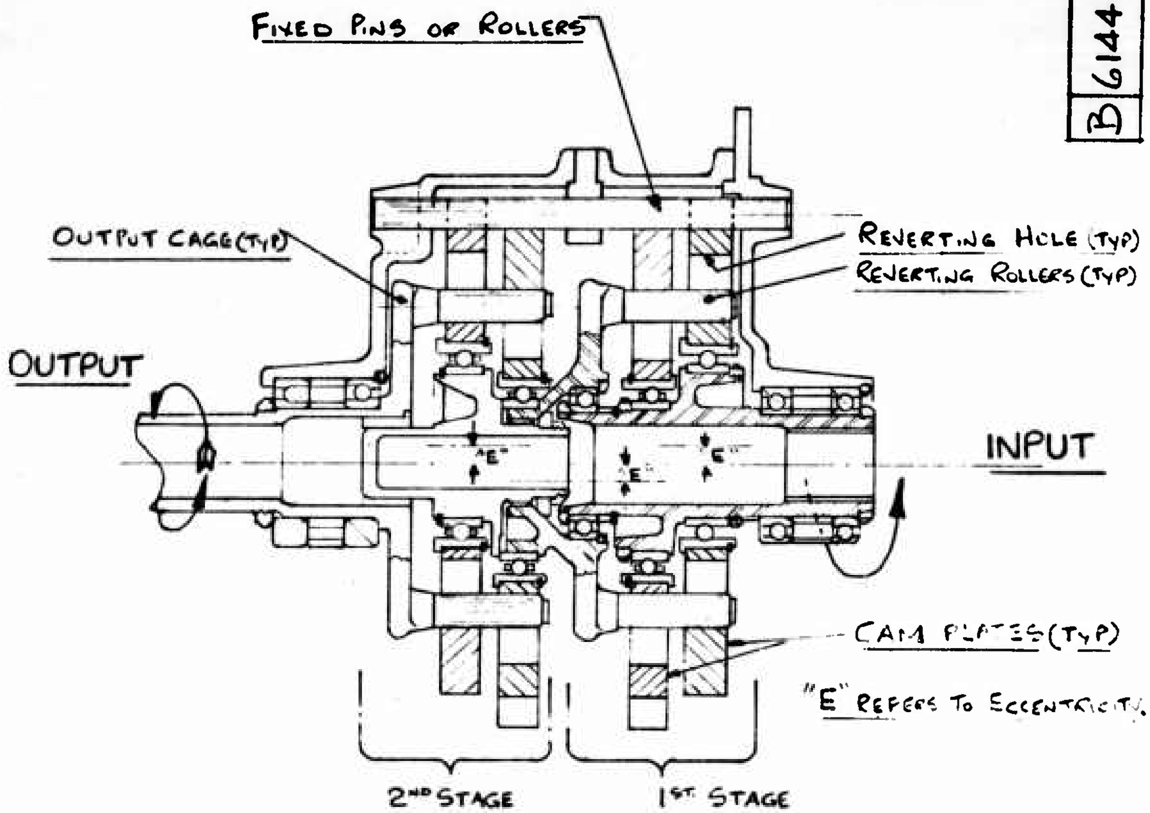
DRAWINGS

WESTERN GEAR CORP. - L :WOOD, CALIF.

BY J. BARROW DATE 7-14-60 SUBJECT CYCLOIDAL CAM DRIVE PRINCIPLE FOR ONE OR TWO STAGES. SHEET NO. 1 OF 1

CHKD. BY PHT DATE 10/29/60 JOB NO. 50584

B 6144-B3.

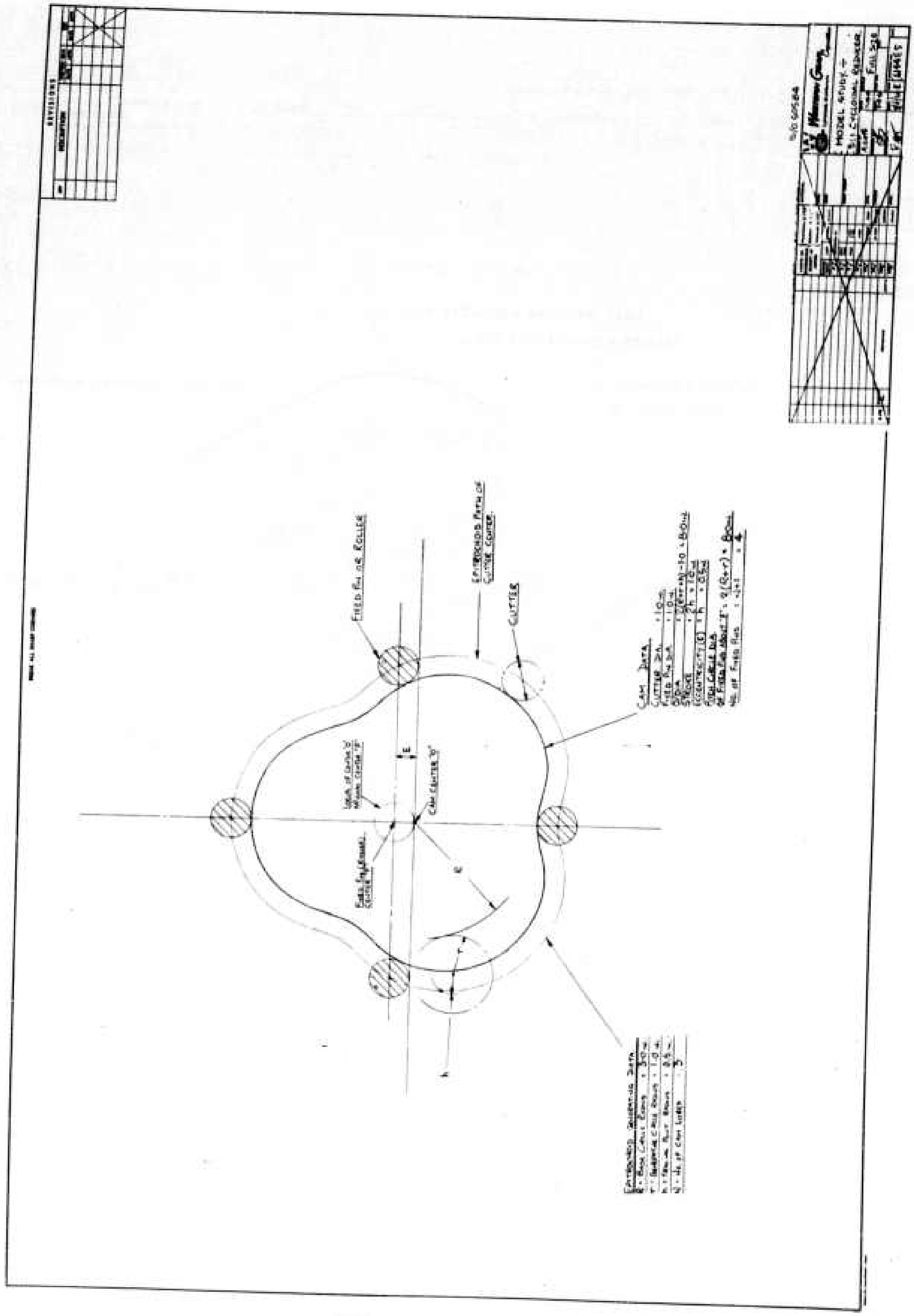


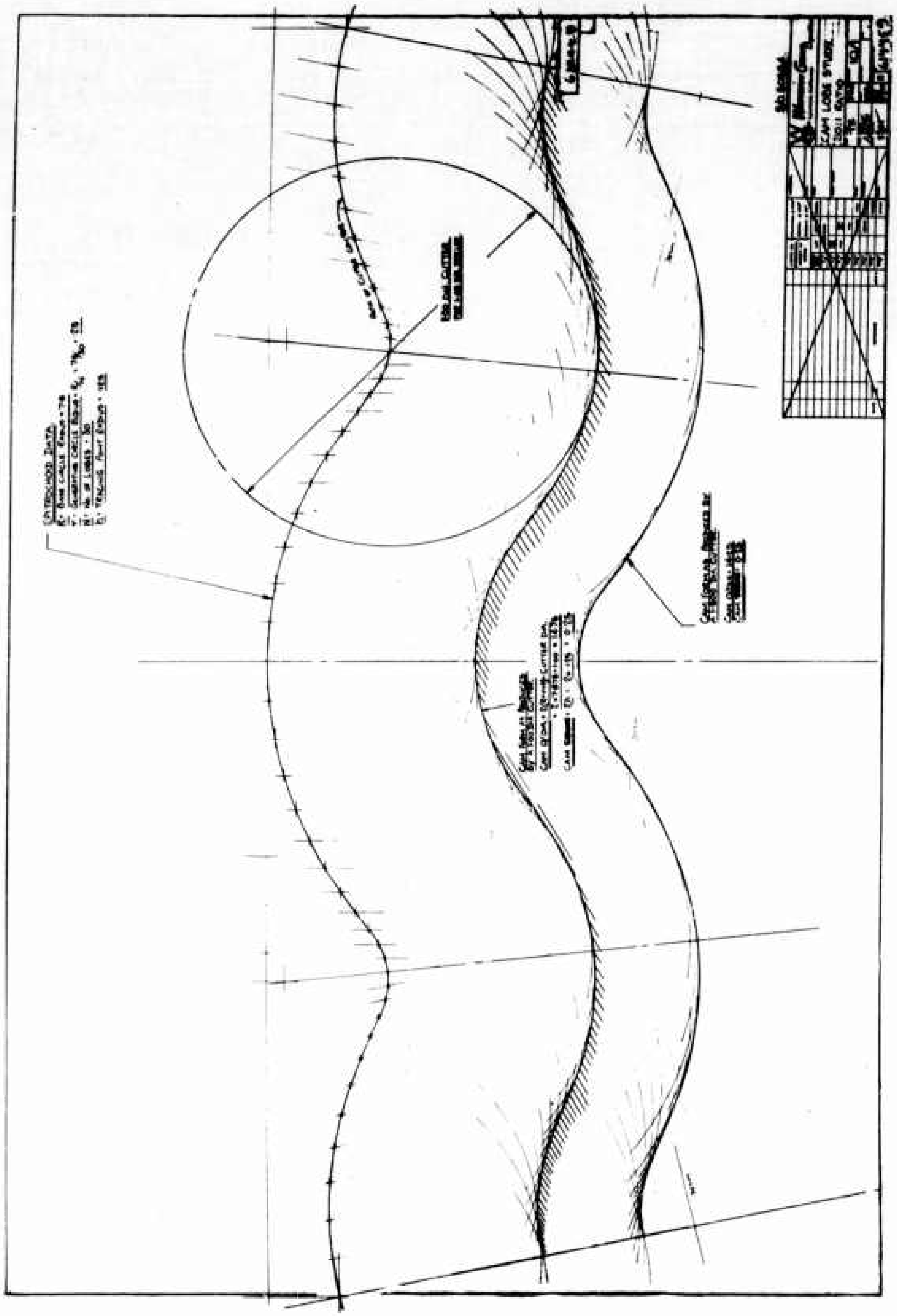
6 LOBES ON CAM = $1/6 \times 1/6 = 1/36$ RED'N ←→ 6 LOBES ON CAM. = $1/6$ REDUCTION

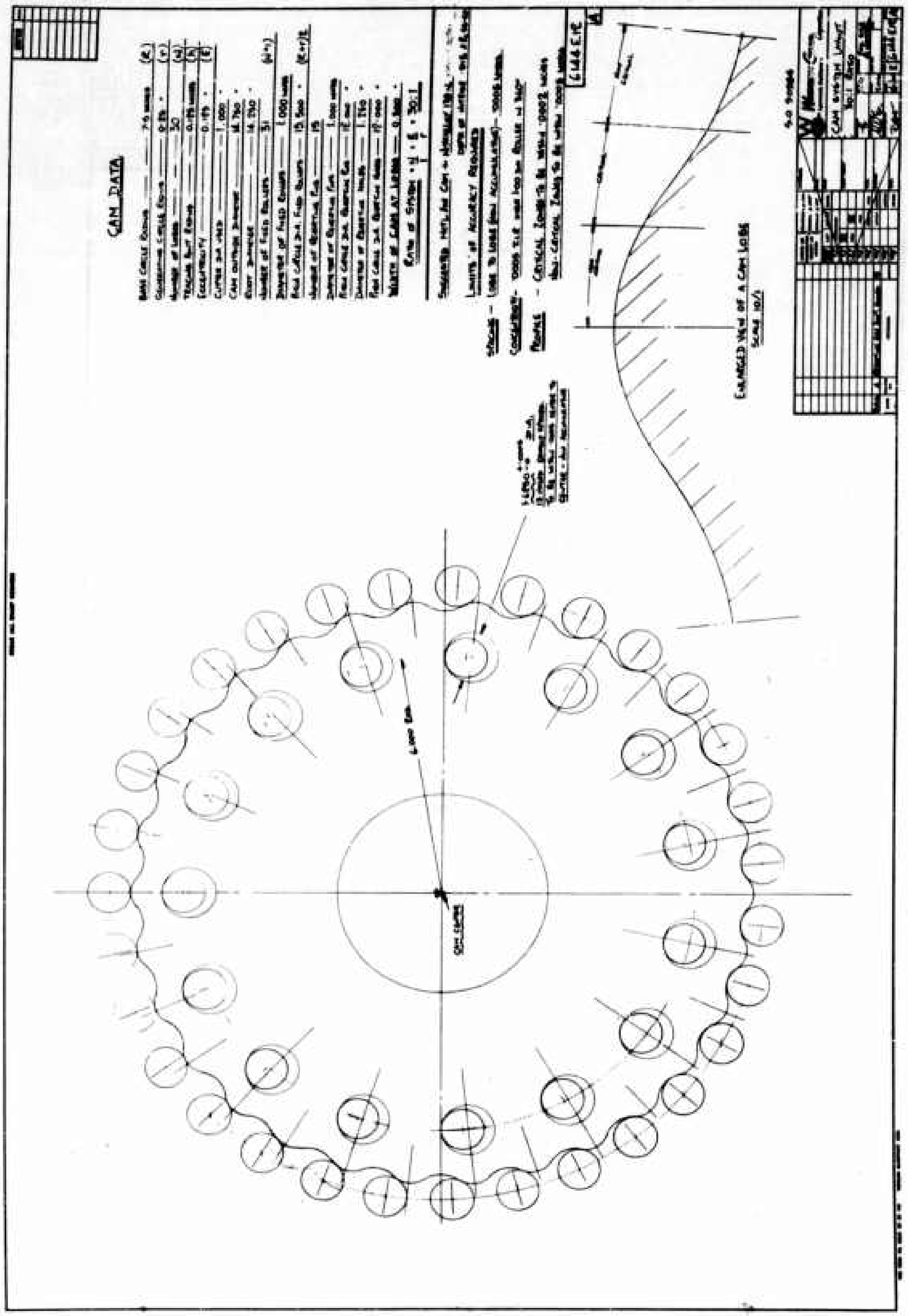
12 " " " = $1/12 \times 1/12 = 1/144$ " ←→ 12 " " " = $1/12$ "

CAM REDUCER SCHEMATIC

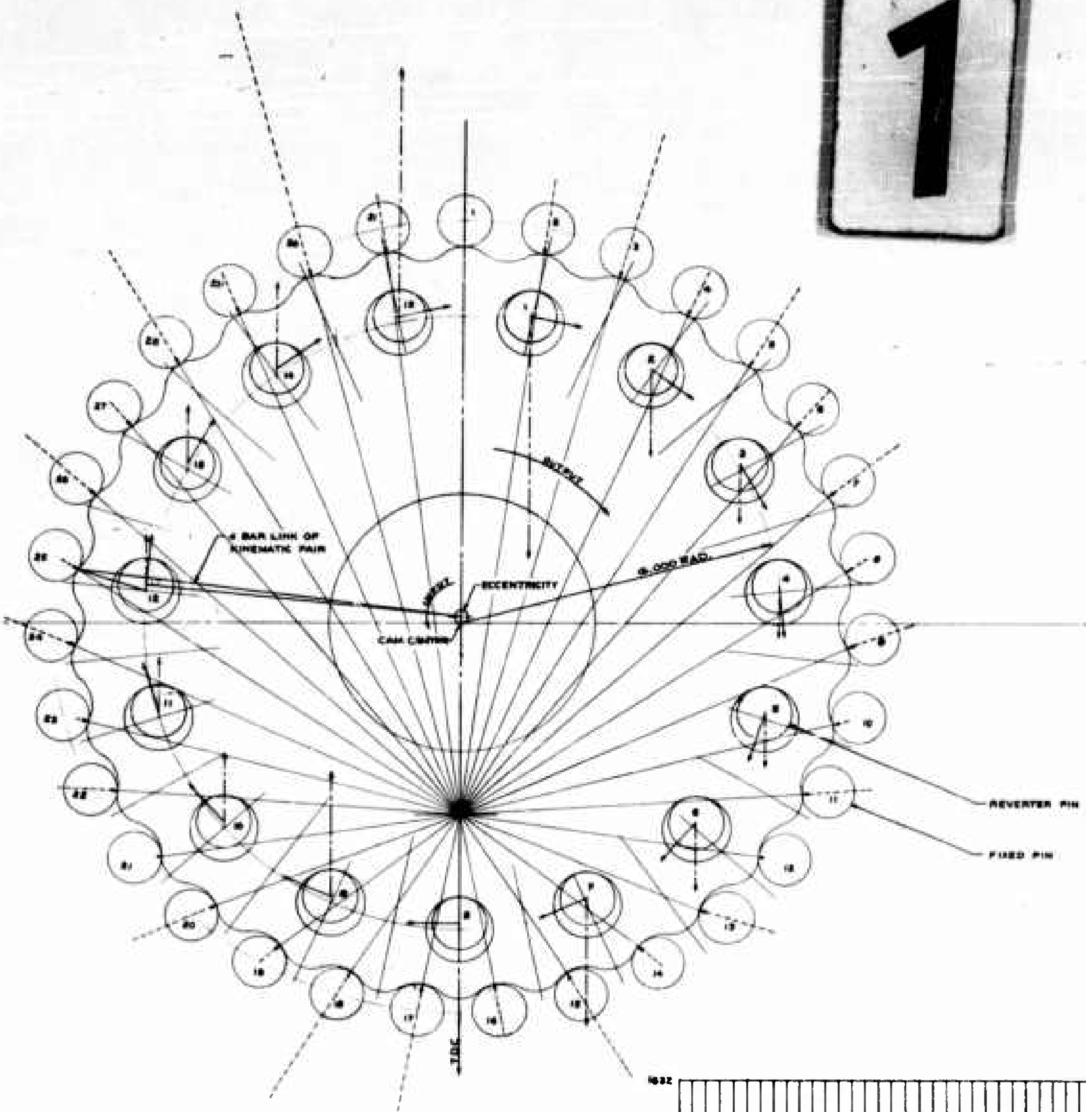
DWG. NO. 6144 B-3.


REVISIONS
FIXED PIN OR ROLLER
CAM CENTER "O"
EPITROCHOID PATH OF CUTTER CENTER
CUTTER
CAM DATA
EPITROCHOID GENERATING DATA
MODEL STUDY
CYCLOIDAL REDUCER
FULL SIZE

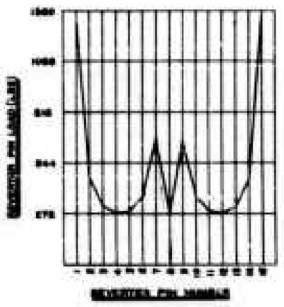



CAM DATA
CAM CENTRE
ENLARGED VIEW OF A CAM LOBE
SCALE 10/1
6144E1E
CAM SYSTEM LAYOUT
30:1 RATIO

1

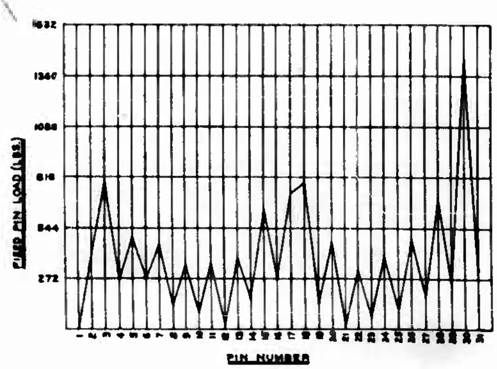


15 OUTPUT PINS
30:1 RATIO

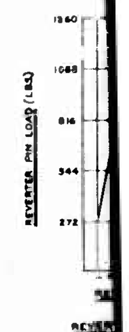


REVERTER PIN LOAD SPECTRUM

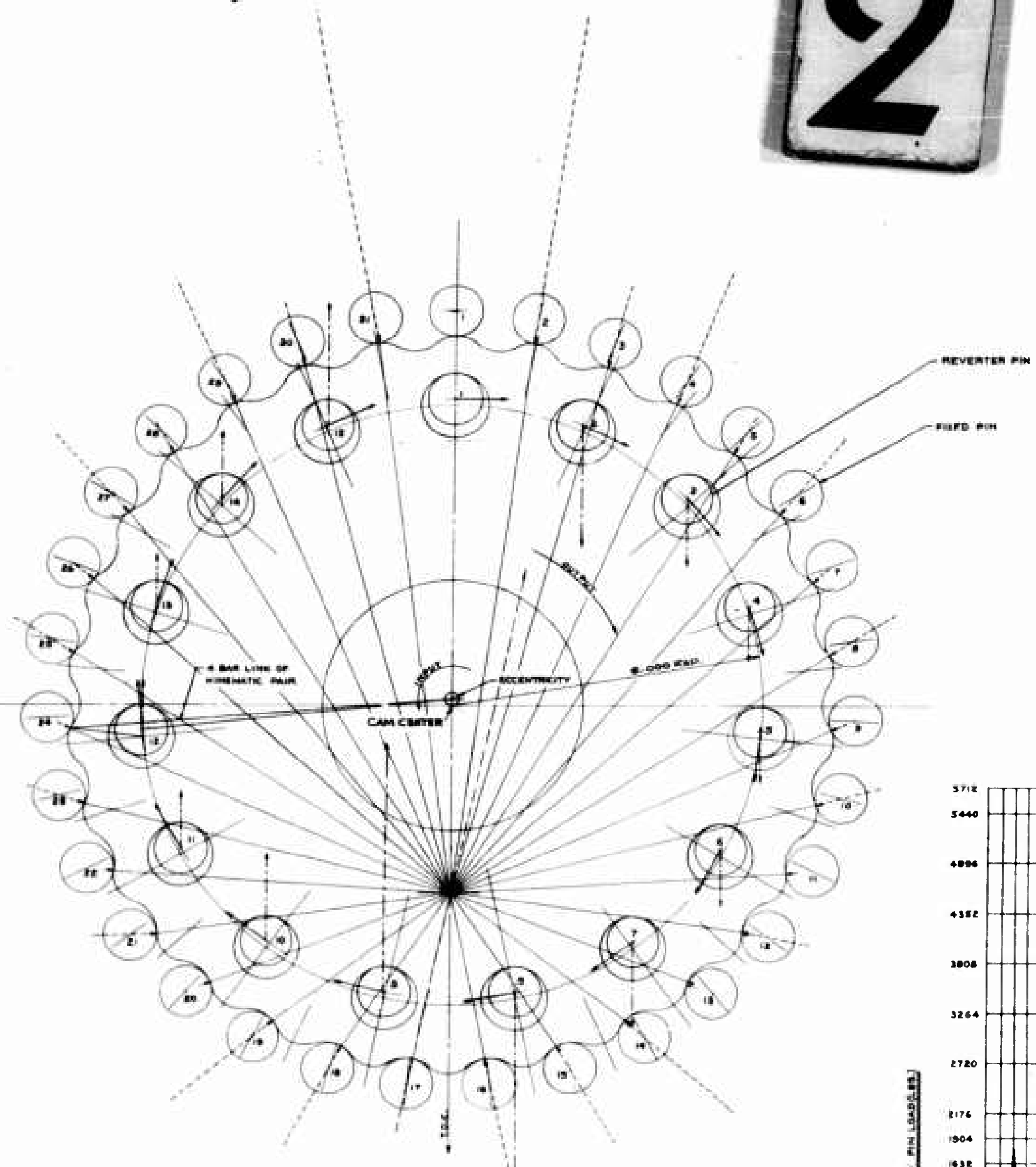
PHASE NO. 1
REVERTER IN LINE
WITH T.D.C. OF CAM

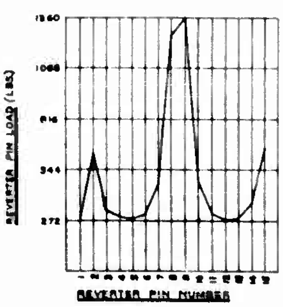


STATIC PIN LOAD SPECTRUM



PUT PINS
RATIO



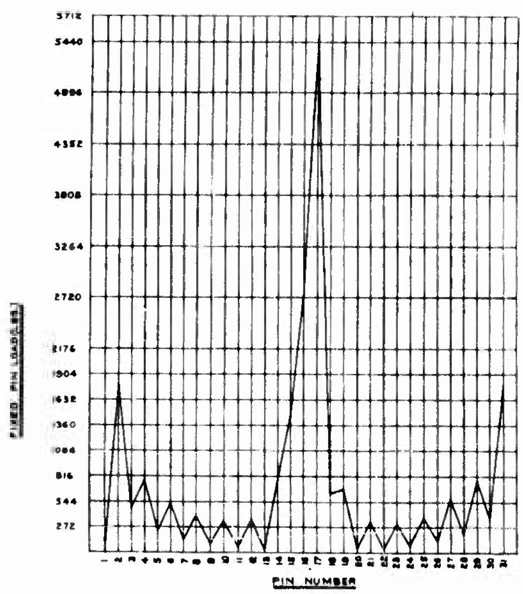


REVERTER PIN LOAD SPECTRUM

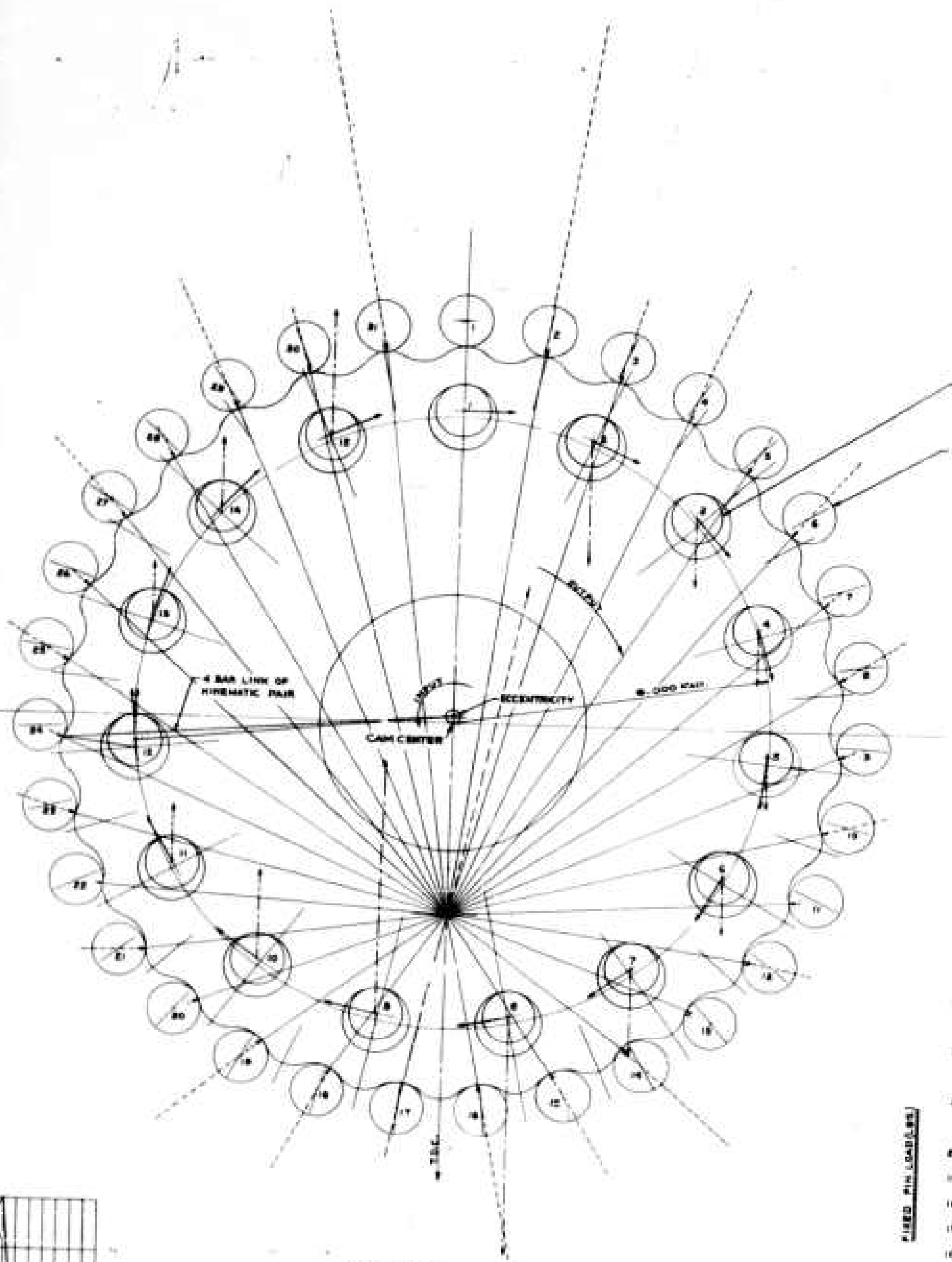
PHASE NO. 2

ADVANCED 1 CAM LOBE FROM PHASE NO.1 REVERTERS STRADDLE ℄ AT T.D.C OF CAM

OUTPUT REVERTER PIN LOAD
RESULTANT REVERTER PIN LOAD
STATIC PIN LOAD
SCALE 272 LBS. PER IN.



STATIC PIN LOAD SPECTRUM

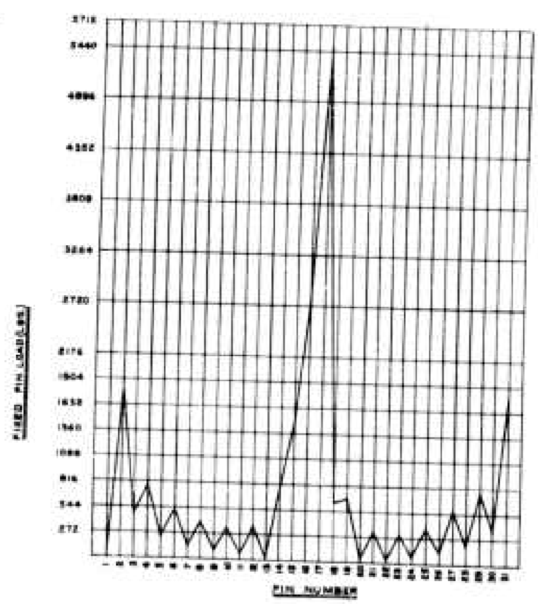


OAD SPECTRUM

PHASE NO. 2

ADVANCED 1 CAM LOBE FROM PHASE NO. 1
REVERTERS STRADDLE ℄ AT T.D.C. OF CAM

——— OUTPUT REVERTER PIN LOAD
— - — RESULTANT REVERTER PIN LOAD
- - - - STATIC PIN LOAD
SCALE 272 LBS. PER. IN



STATIC PIN LOAD SPECTRUM

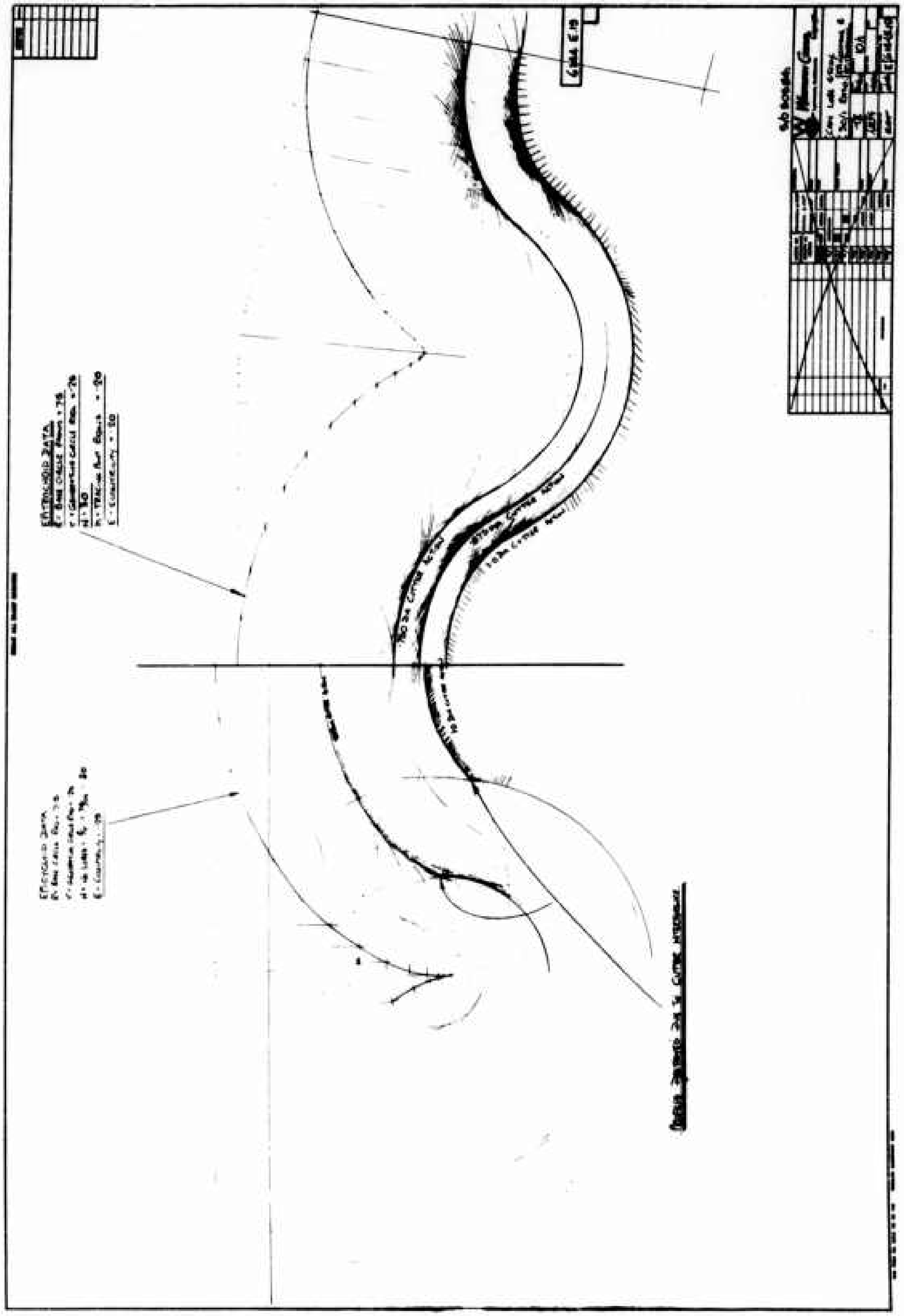
SO 80584

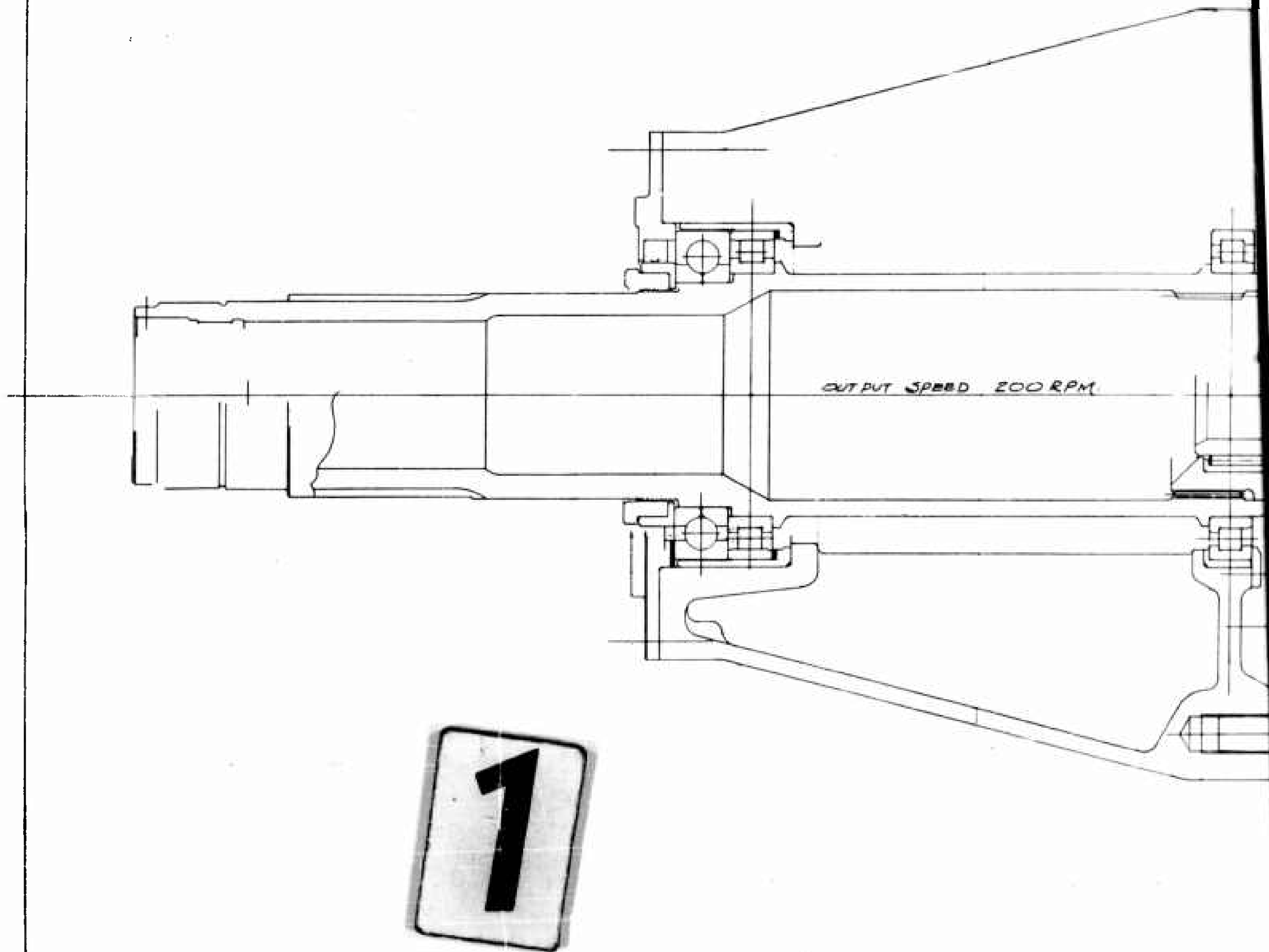
Western Gear Corporation

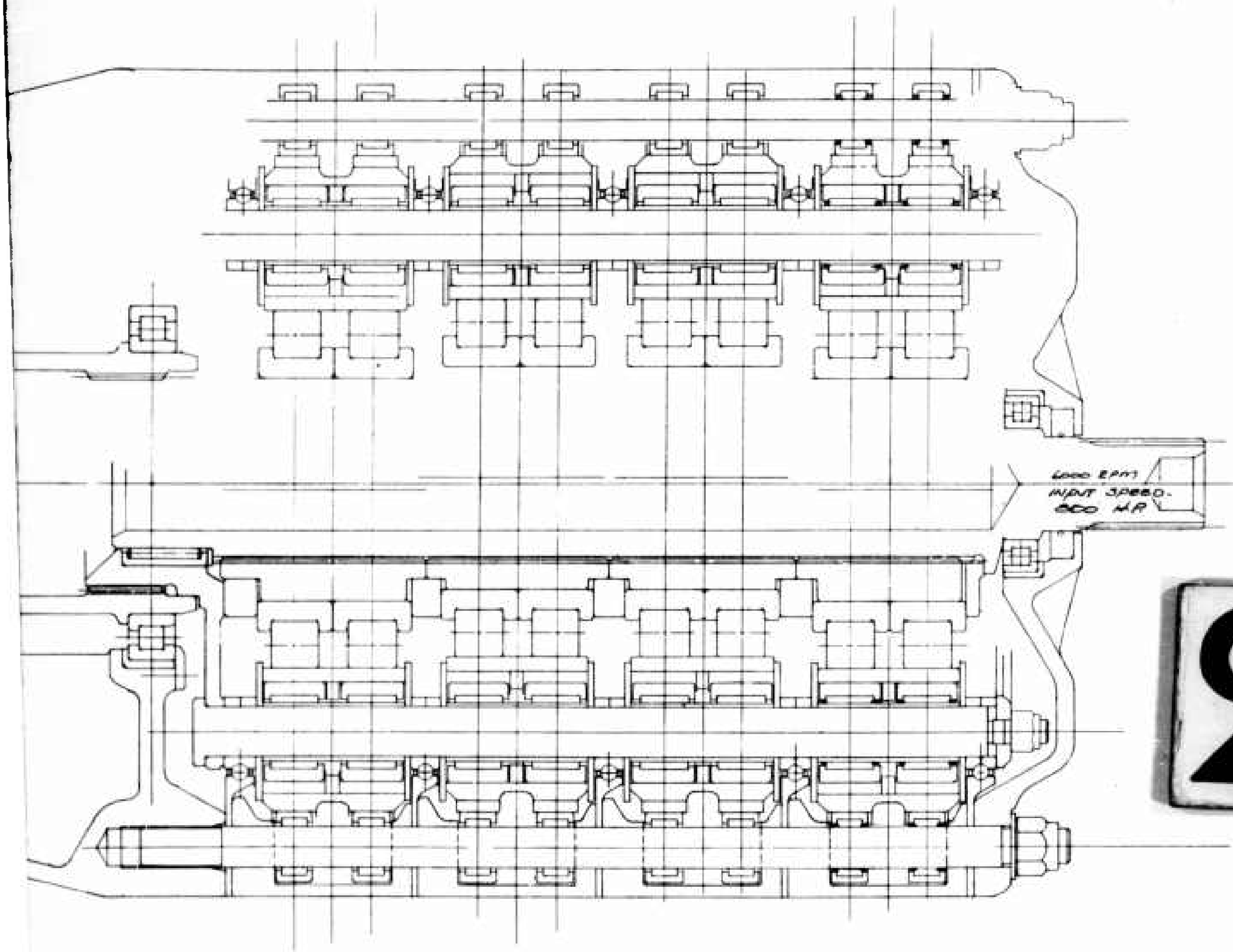
CAM SYSTEM
LOAD PATTERN

FULL SIZE

5144817

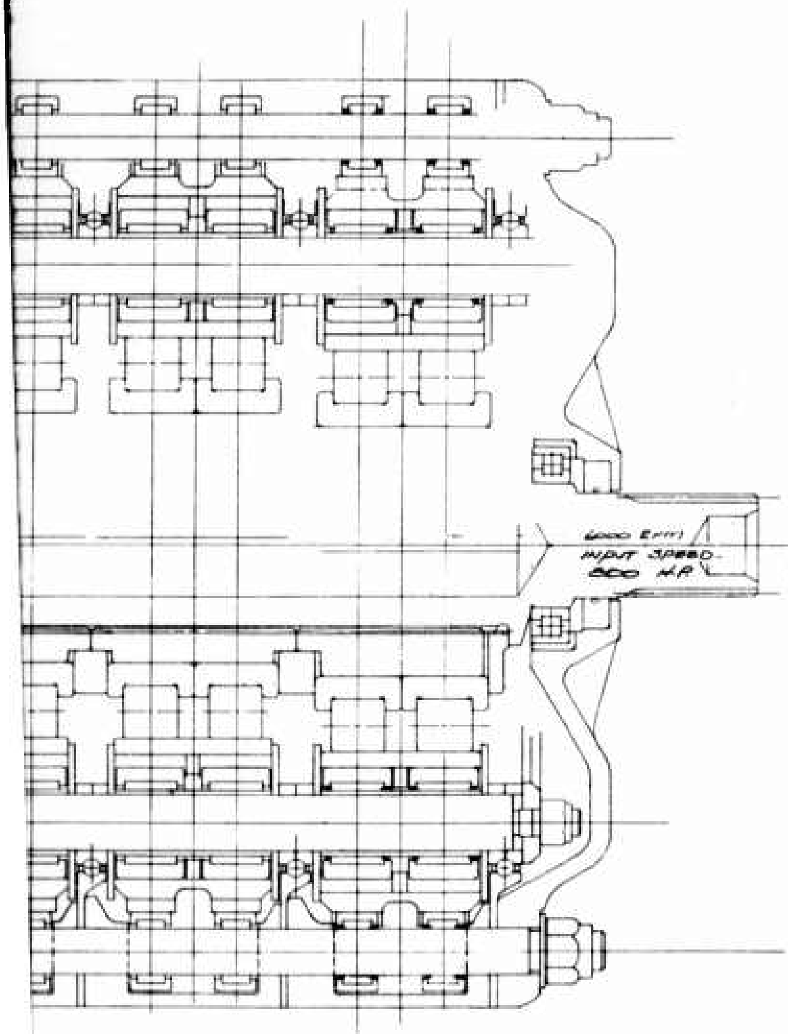



OUTPUT SPEED 200 RPM
1



2

S.O. 5055



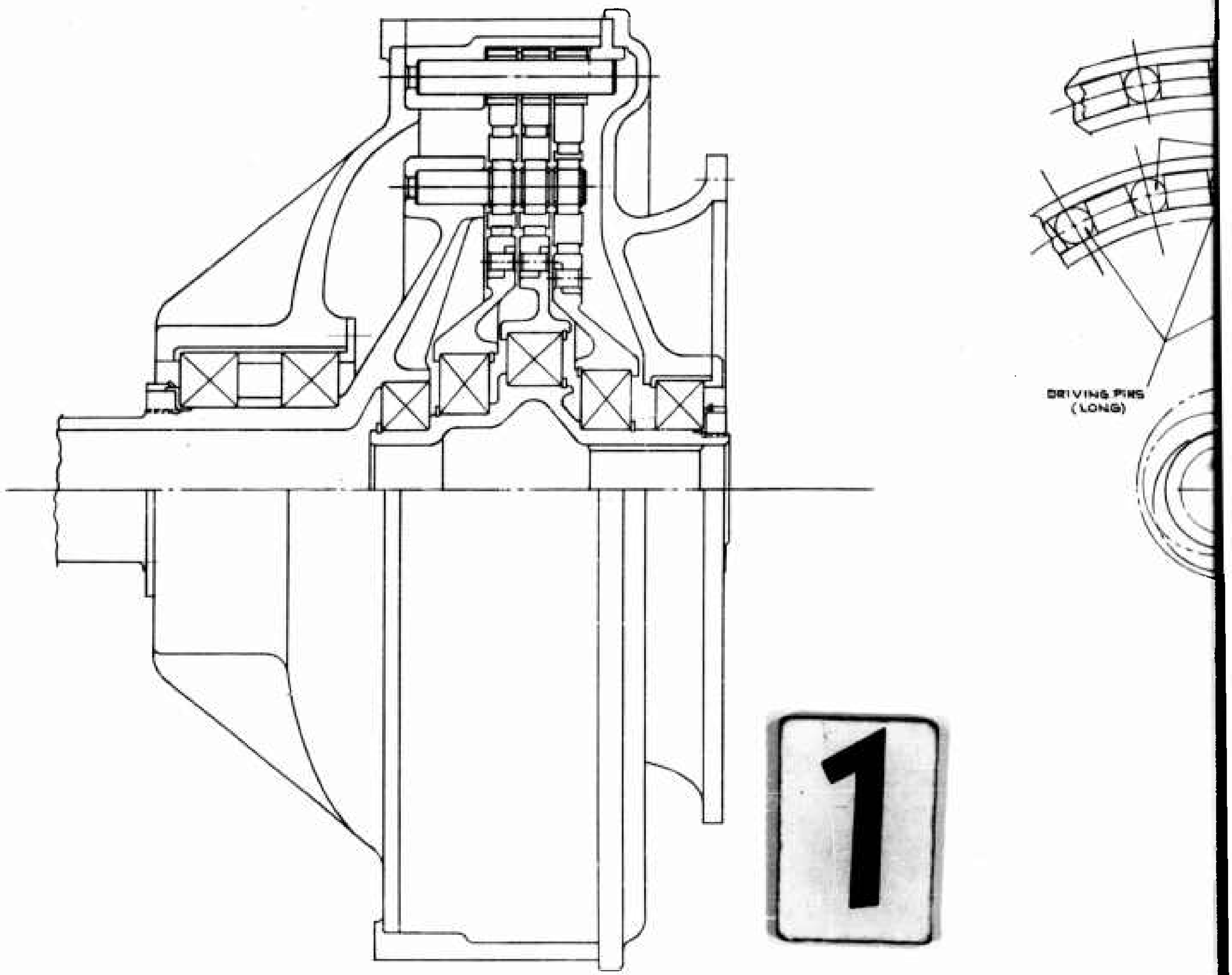
3

S.O. 50564

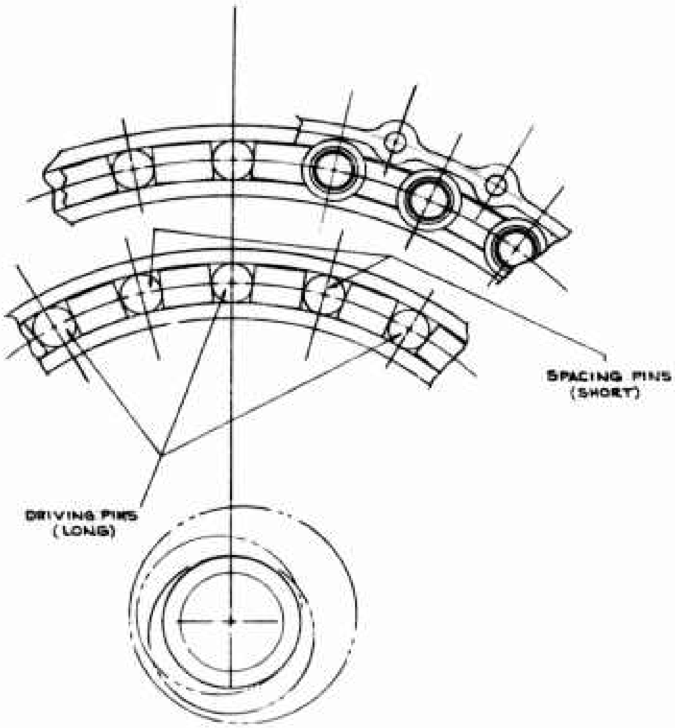
Western Gear

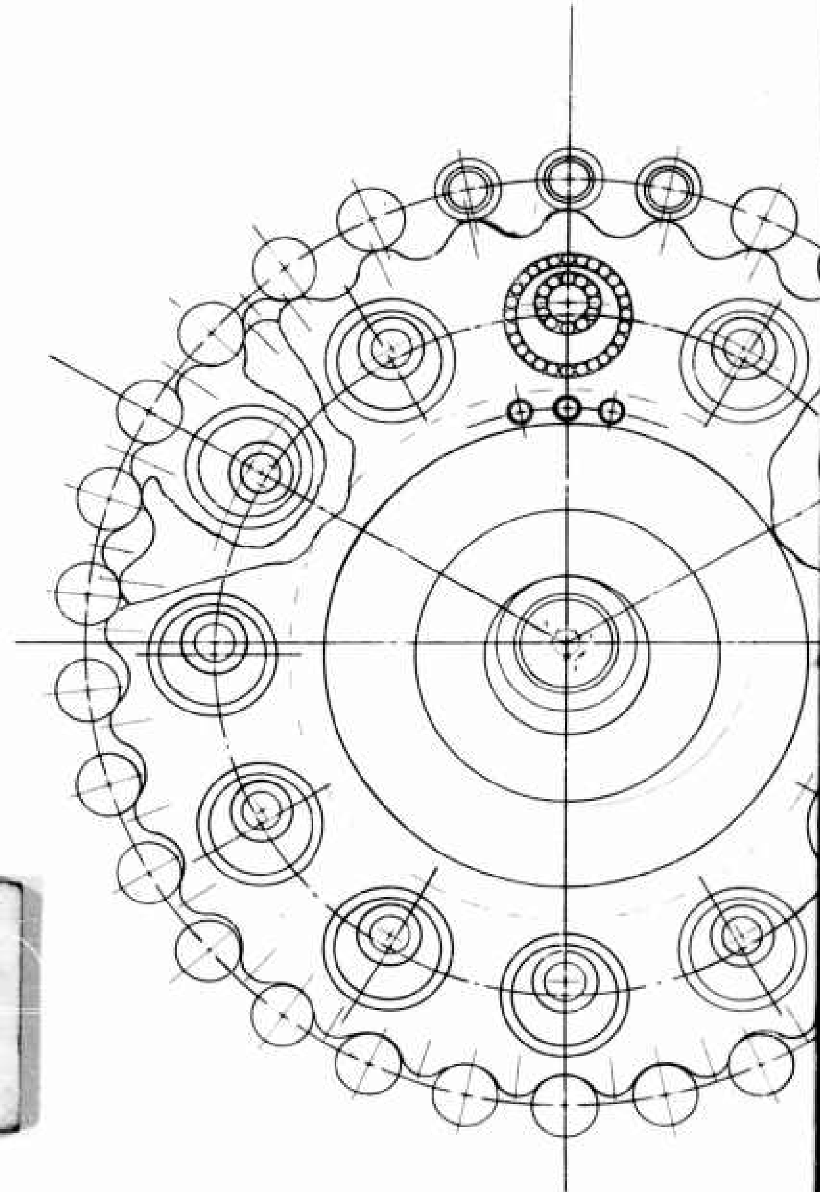
PRELIMINARY LAYOUT

180:1 REDUCTION

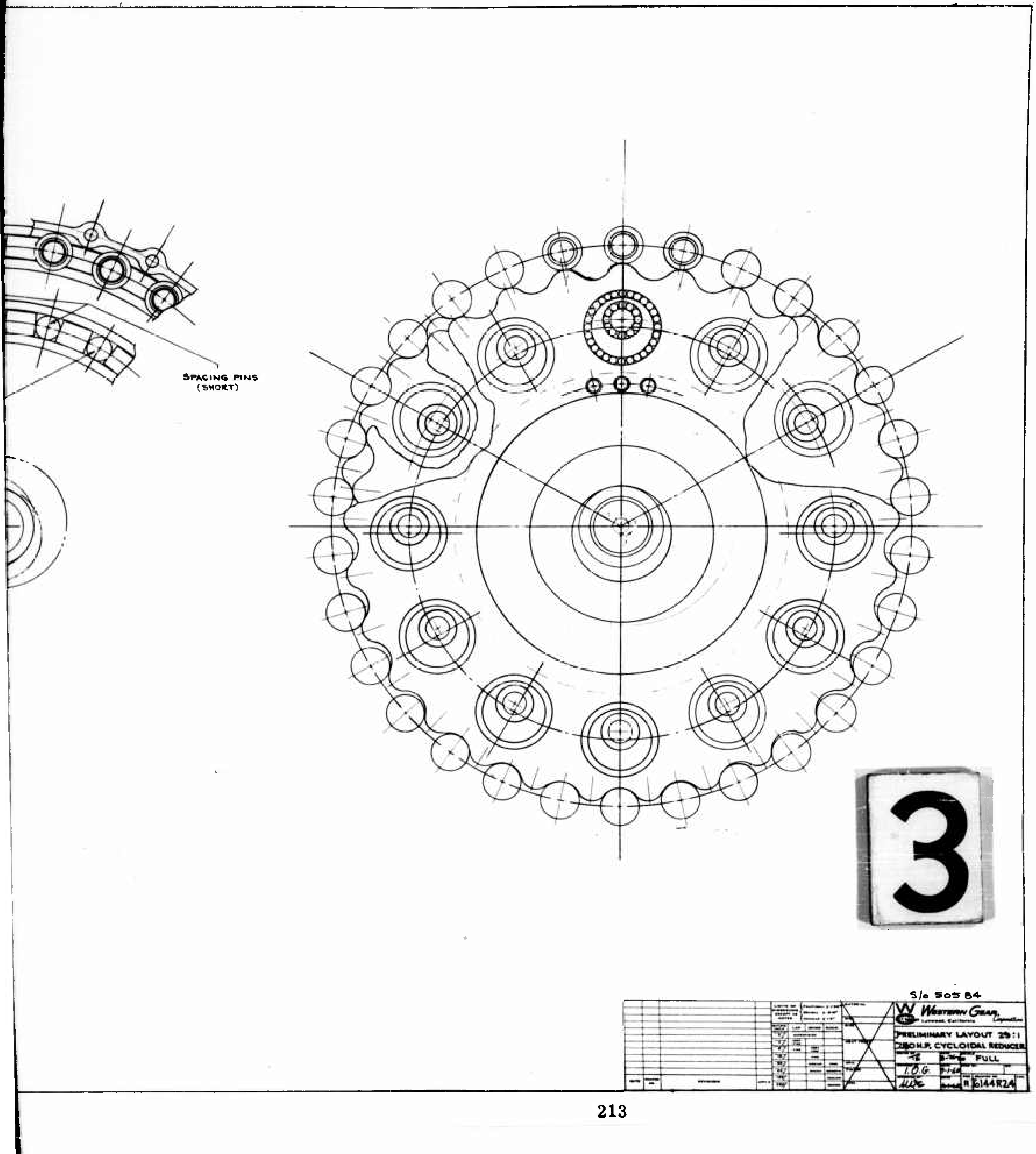


1





2


SPACING PINS
(SHORT)
3
S/o 50584
WESTERN GEAR
PRELIMINARY LAYOUT 29:1
H.P. CYCLOIDAL REDUCER
FULL
I.O.G.
R 6144R24

WESTERN GEAR CORP. - LYNWOOD, CALIF.

BY T. BARROW DATE 9-23-60 SUBJECT CYCLOIDAL CAM TRANSMISSION, MODEL ASSEMBLY SHEET NO. 1 OF 1

CHKD. BY RM DATE 10-5-60 JOB NO. 50584

DATE	BY	CHG. NO	REVISIONS
10/3/60	TB	A	WOODEN COVER REPLACED BY LUCITE. SNAP RING CLIPS REPLACE SCREWS
10/5/60	TB	B	DIALS REMOVED.
10/[illegible]	TB	C	[illegible] DRIVE DOWEL & [illegible]

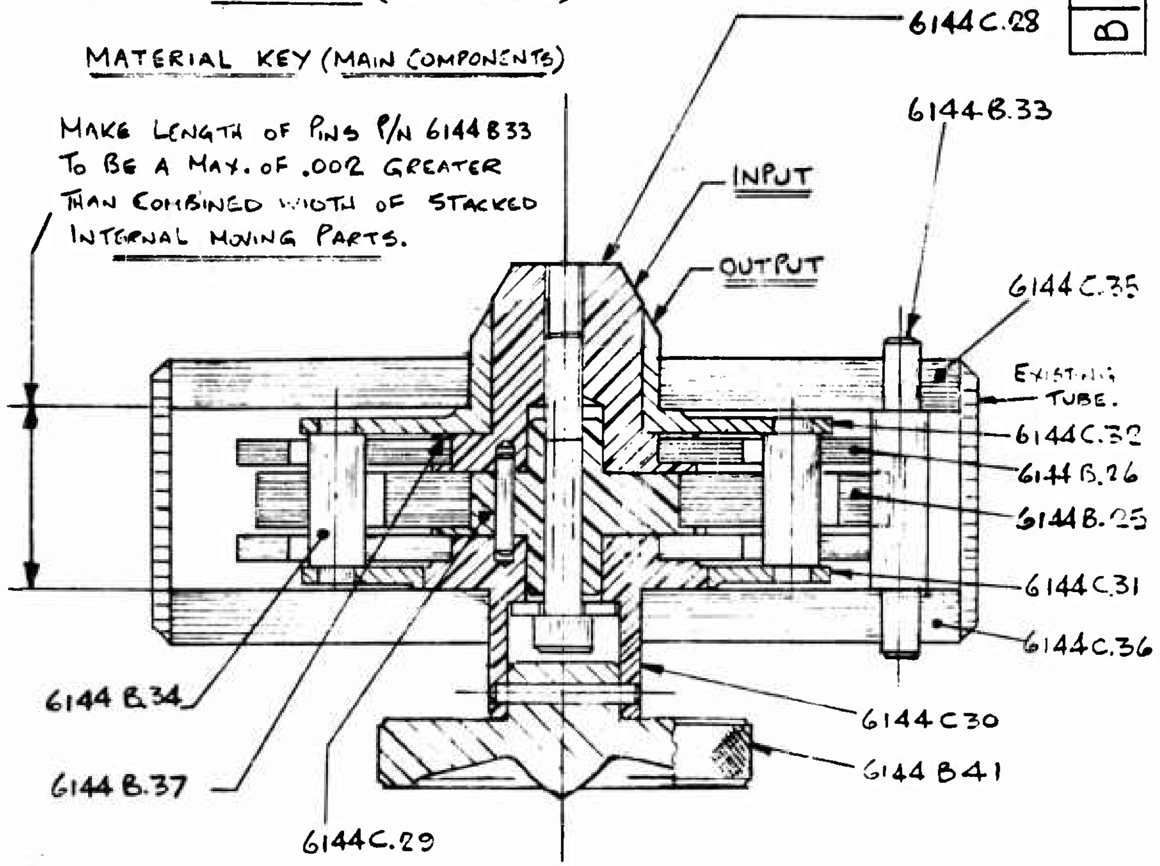
B | 6144 B27 | C

LUCITE

ALUM ALLOY ("GOLD" ANODIZE)

ALUM ALLOY (GREEN ANODIZE)

MATERIAL KEY (MAIN COMPONENTS)

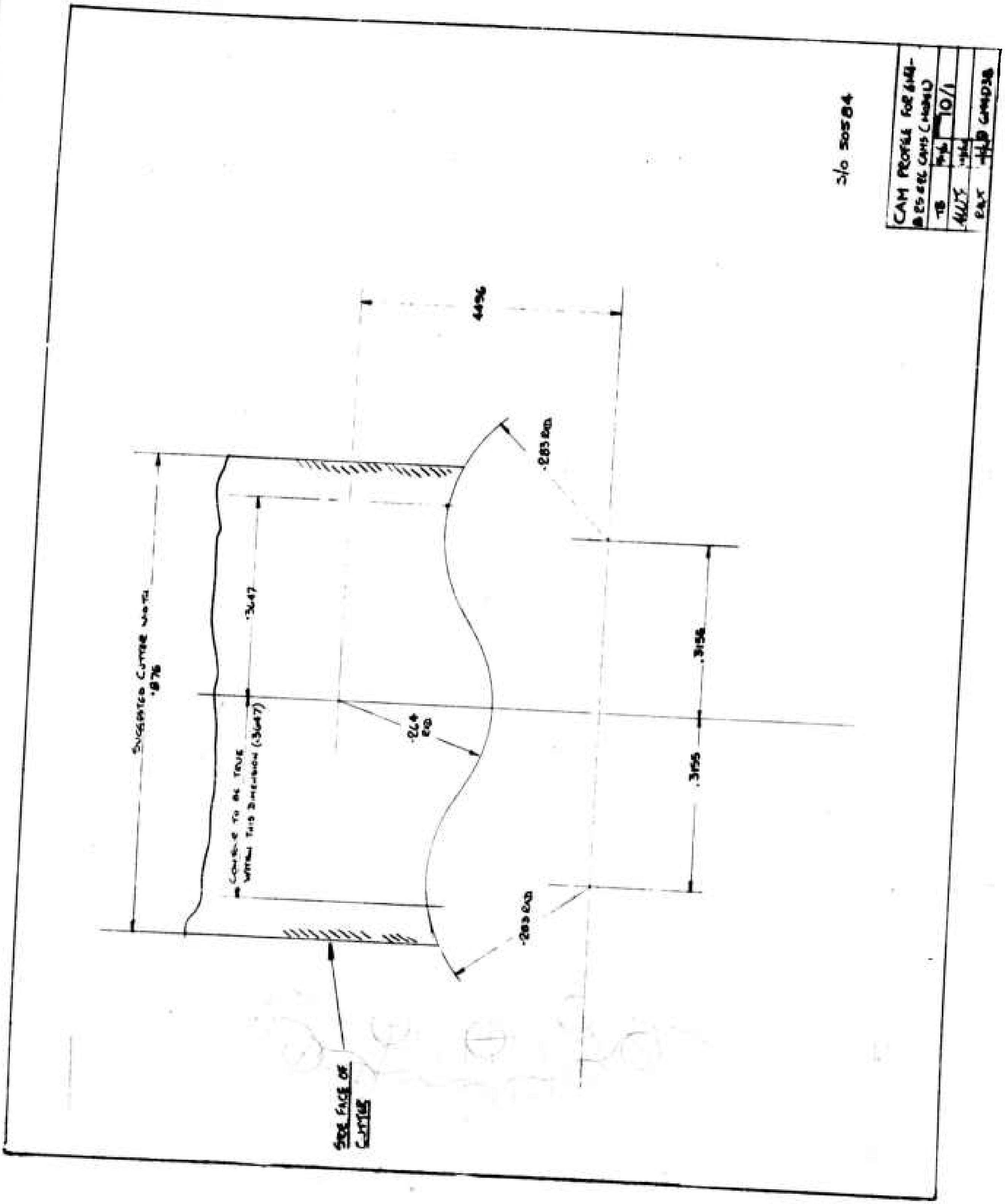


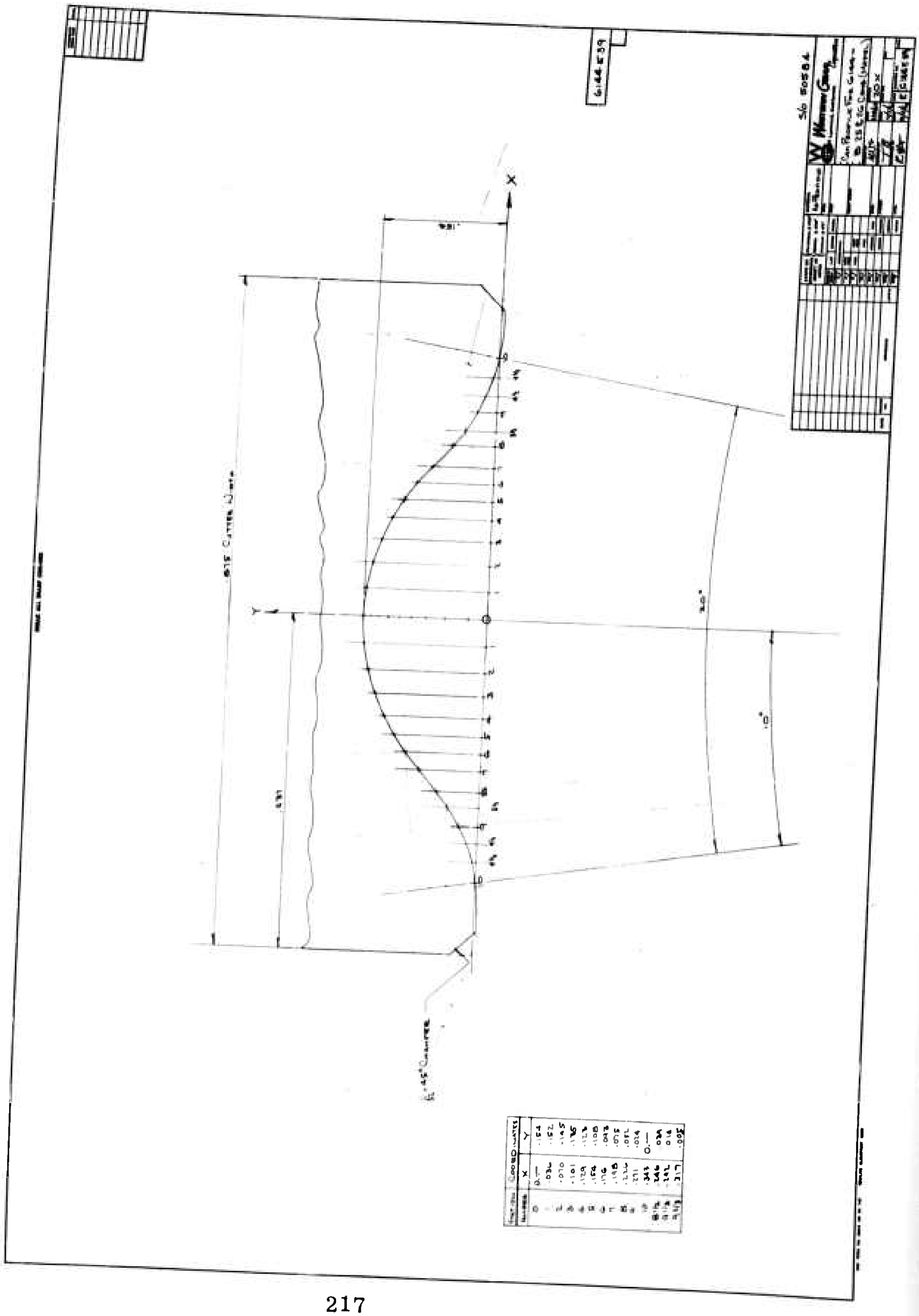
ASSEMBLY - CYCLOIDAL CAM REDUCER MODEL.

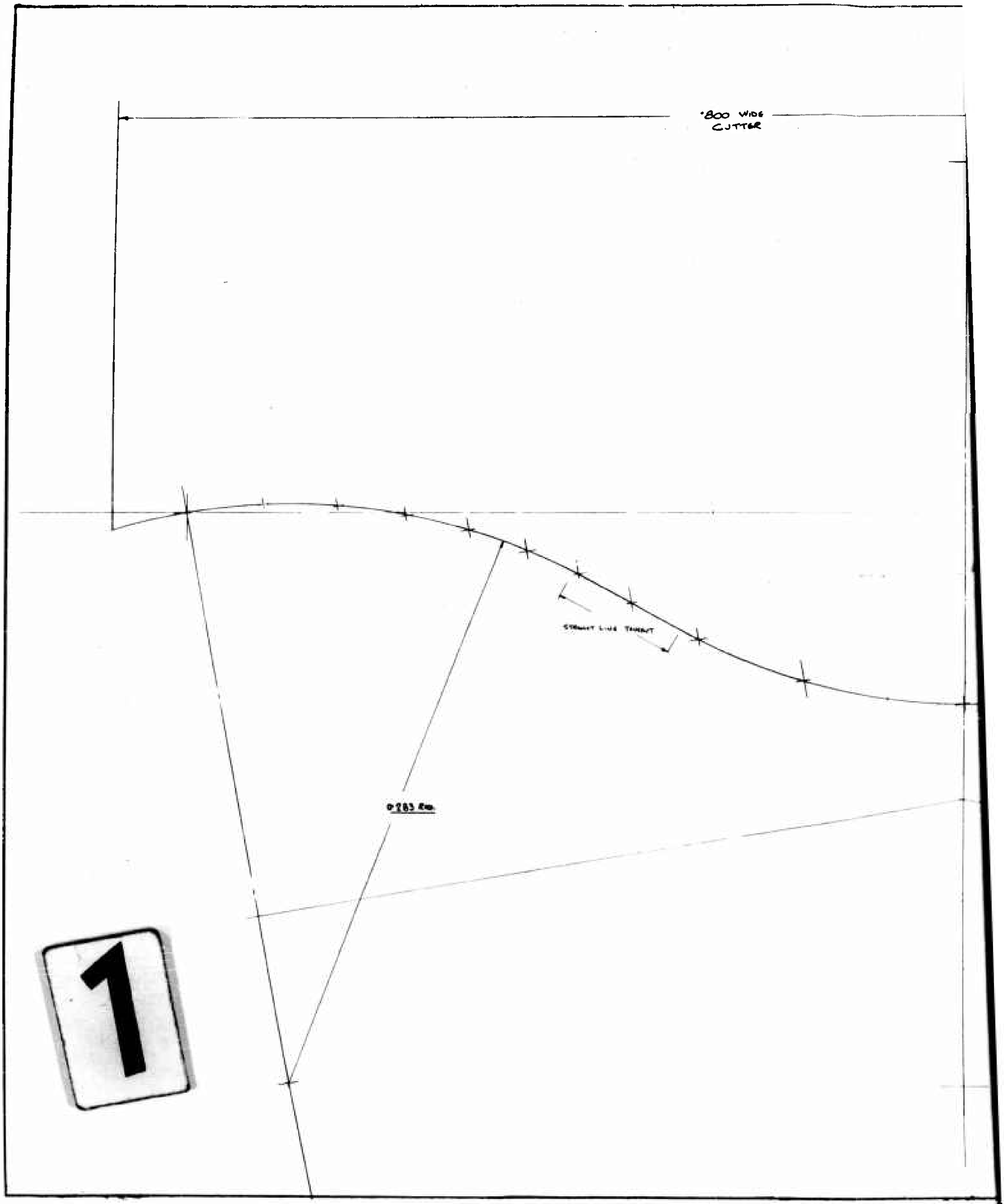
RATIO = 18/1 (SINGLE STAGE)

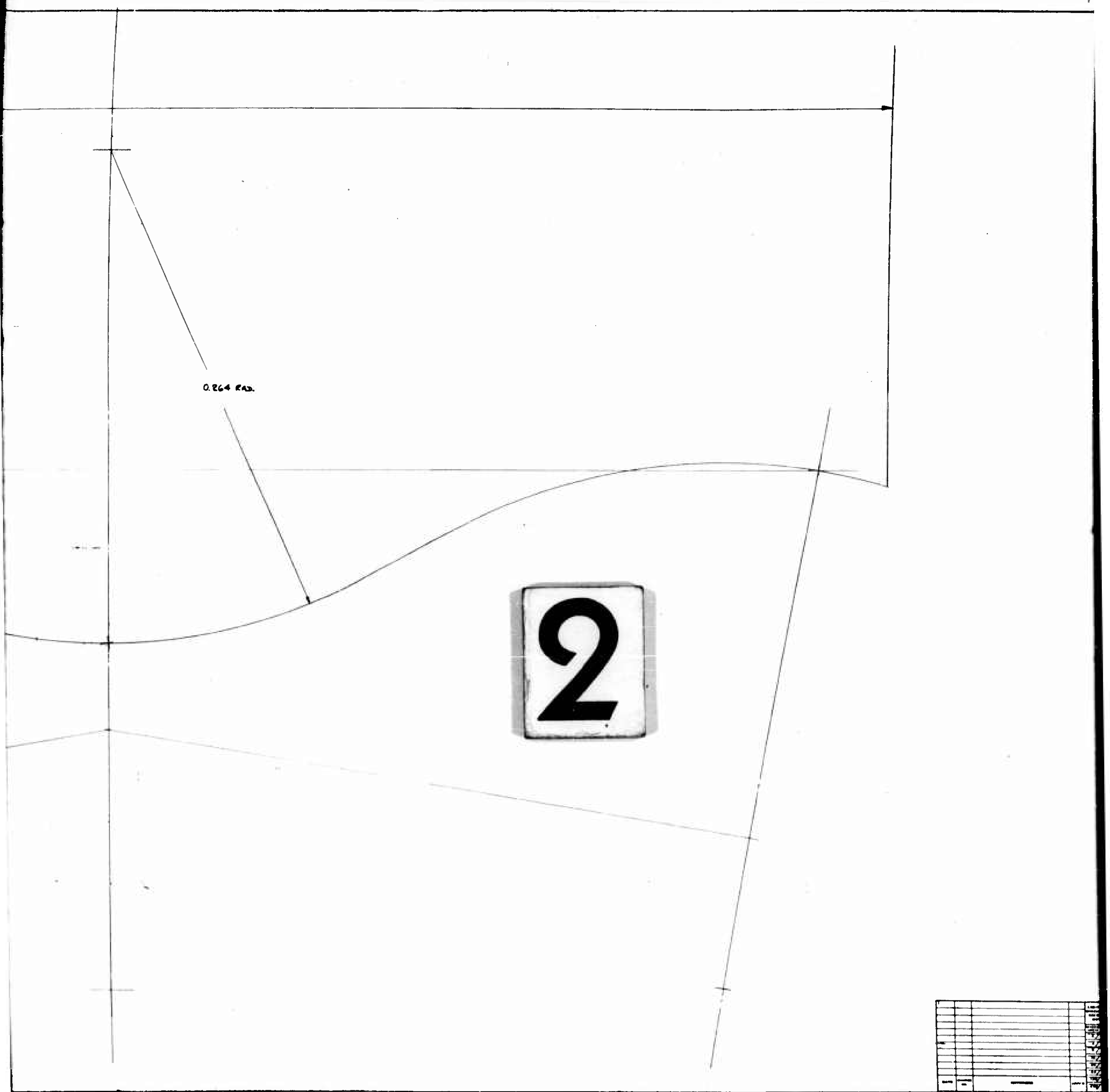
SCALE = FULL SIZE

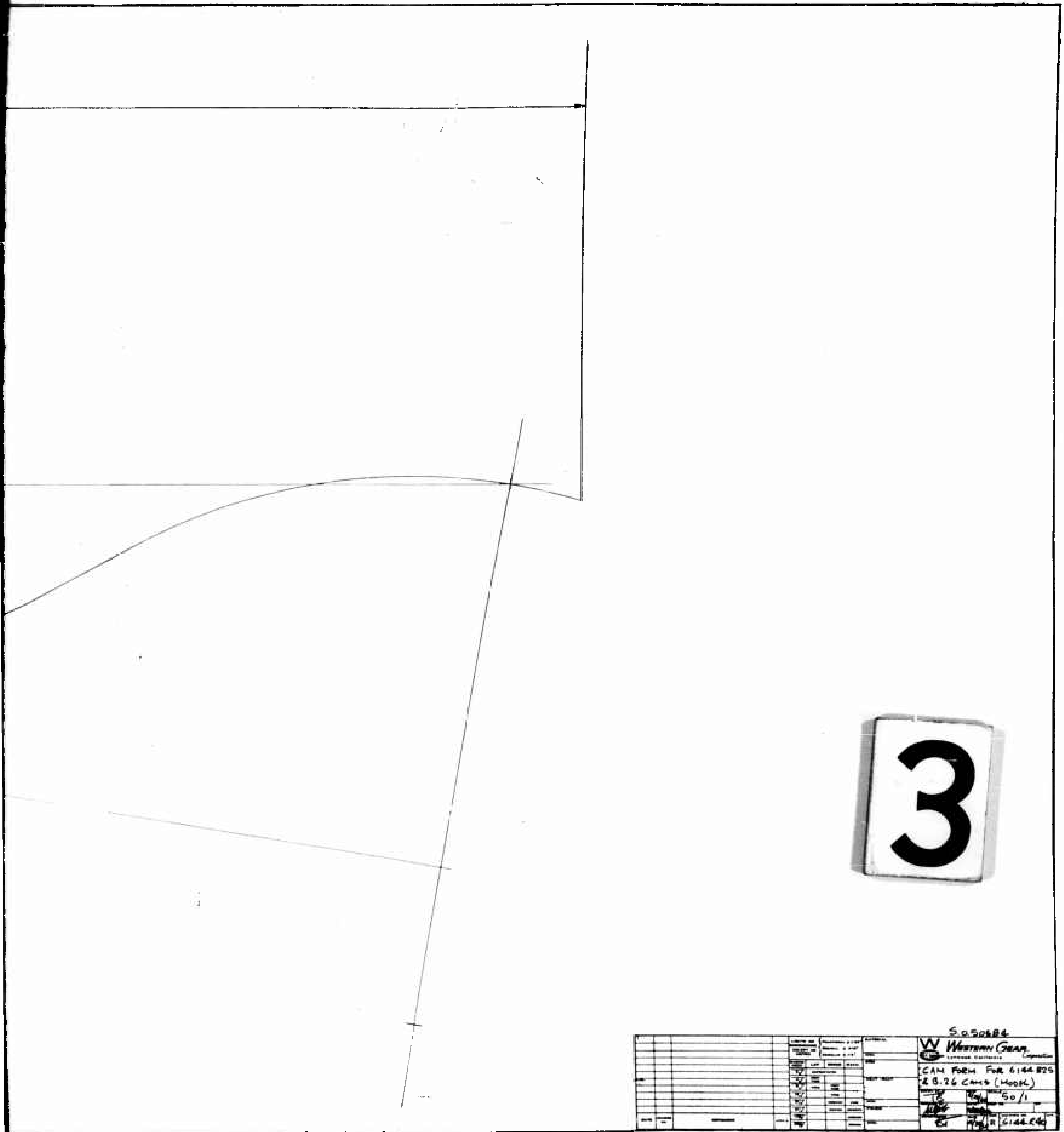
B	6144 B27	C

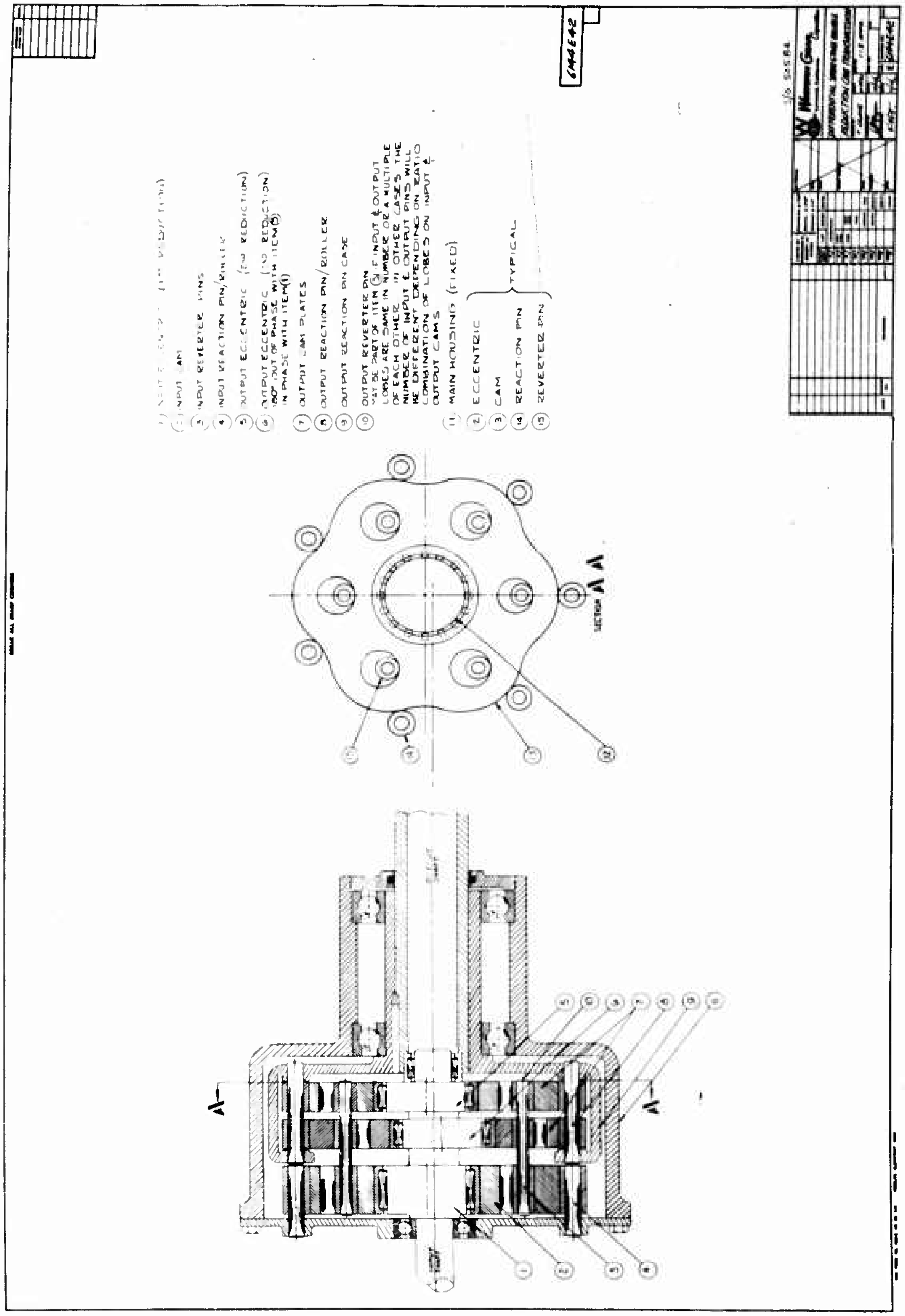

CAM PROFILE
SUGGESTED CUTTER WIDTH
.876
.3647
.3156
.3155
.264 RAD
.283 RAD
.283 RAD
SIDE FACE OF CUTTER
10/1

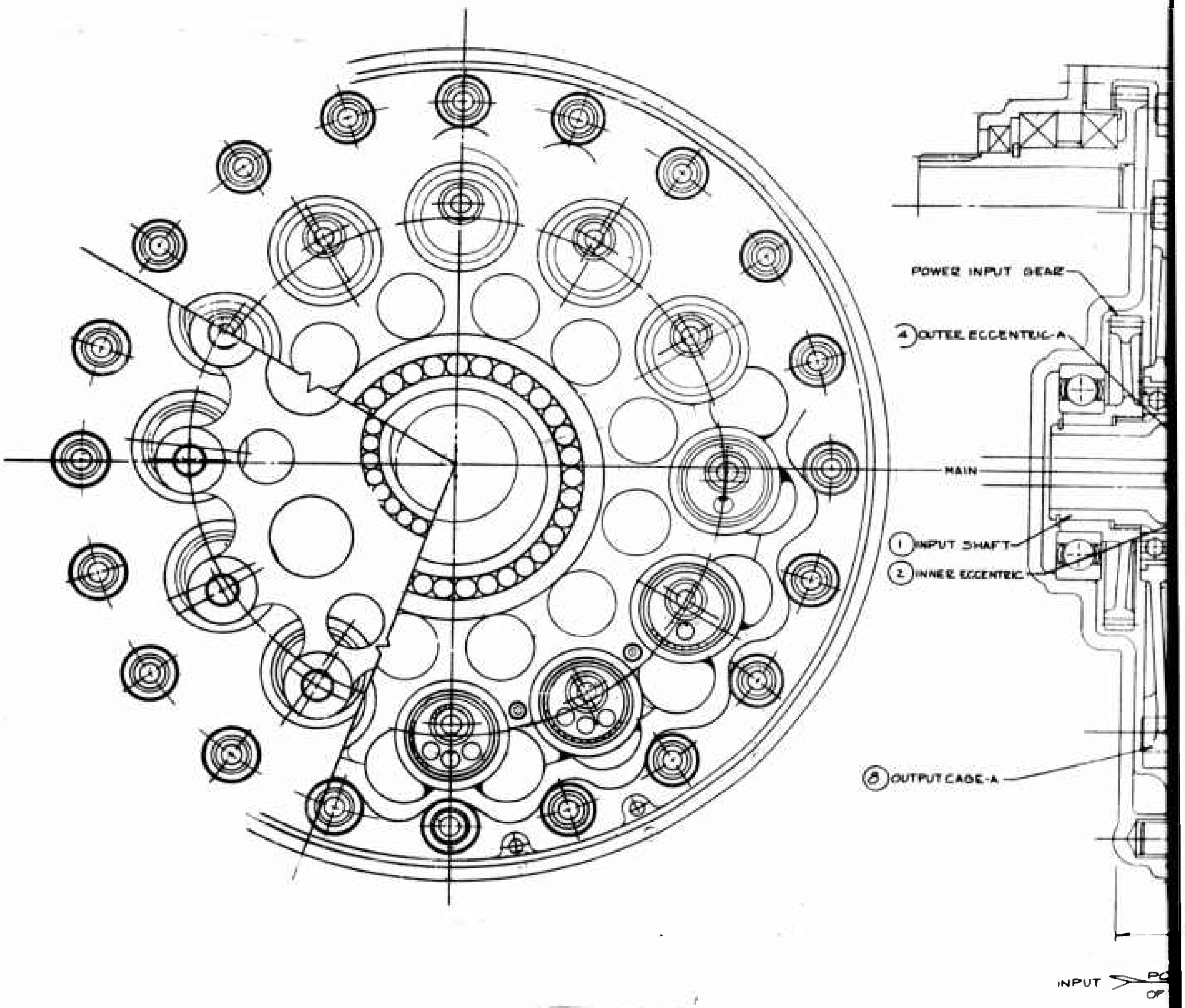



.800 WIDE
CUTTER
1

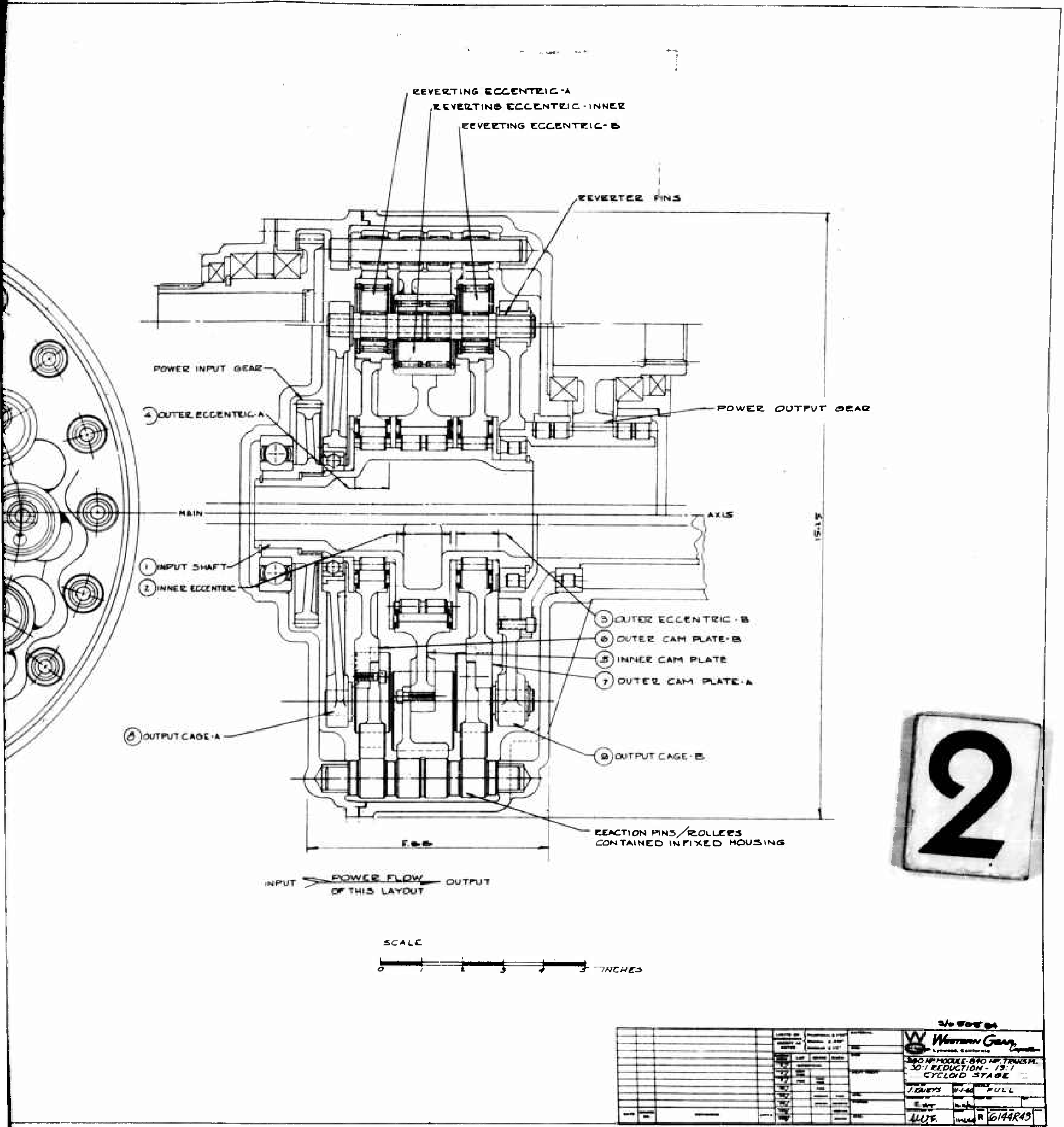

0.264 RAD.
2


3
Western Gear Corporation
CAM FORM FOR 6144 825 & 8.26 CAMS (HOOK)
50/1
6144 C40

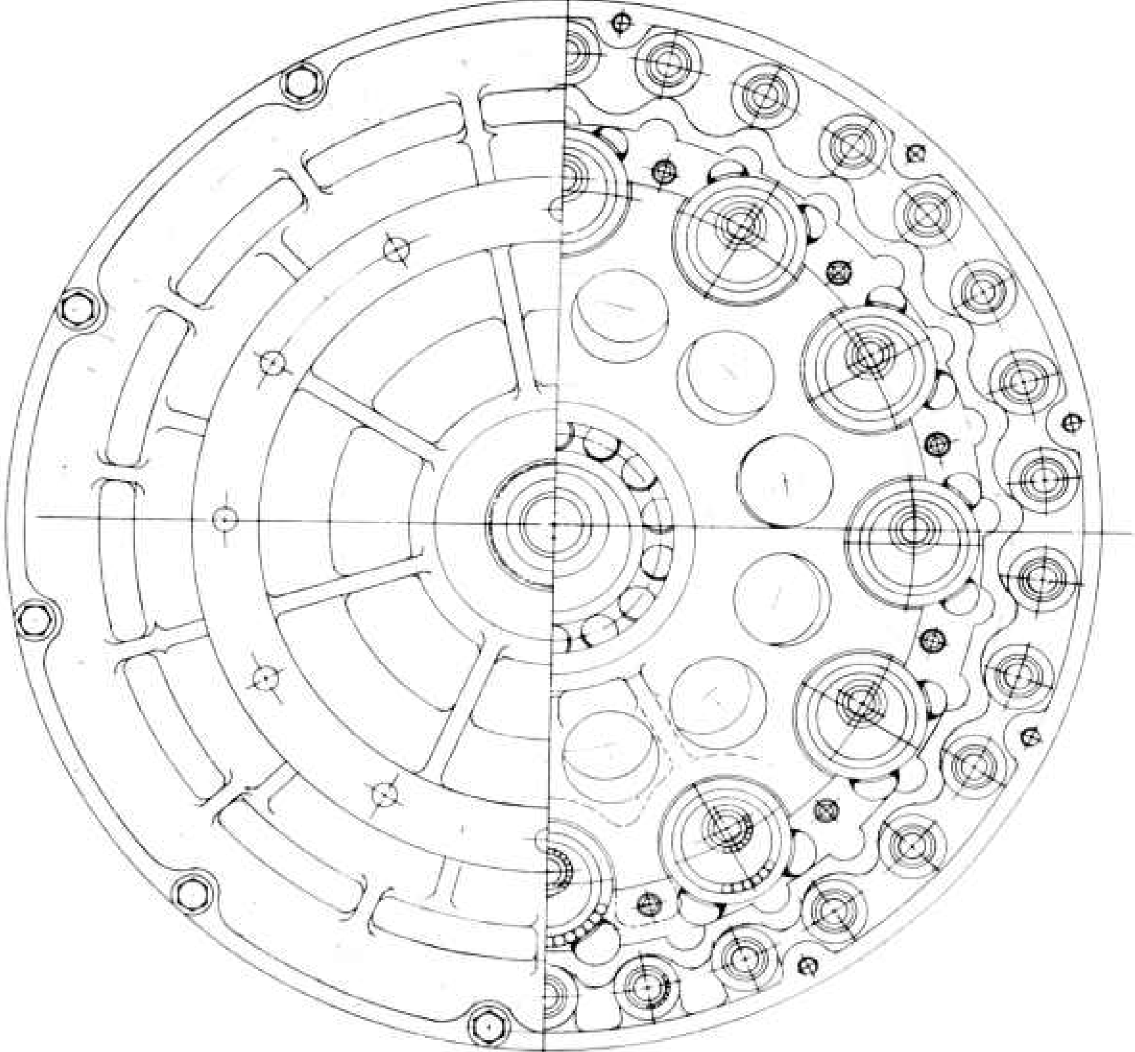

2 INPUT CAM
3 INPUT REVERTER PINS
4 INPUT REACTION PIN/ROLLER
5 OUTPUT ECCENTRIC (2ND REDUCTION)
6 OUTPUT ECCENTRIC (2ND REDUCTION) 180° OUT OF PHASE WITH ITEM (5) IN PHASE WITH ITEM (1)
7 OUTPUT CAM PLATES
8 OUTPUT REACTION PIN/ROLLER
9 OUTPUT REACTION PIN CASE
10 OUTPUT REVERTER PIN MAY BE PART OF ITEM (3) IF INPUT & OUTPUT LOBES ARE SAME IN NUMBER OR A MULTIPLE OF EACH OTHER. IN OTHER CASES THE NUMBER OF INPUT & OUTPUT PINS WILL BE DIFFERENT DEPENDING ON RATIO COMBINATION OF LOBES ON INPUT & OUTPUT CAMS
11 MAIN HOUSING (FIXED)
12 ECCENTRIC
13 CAM
14 REACTION PIN
15 REVERTER PIN
TYPICAL
SECTION A A
6M4E42


POWER INPUT GEAR
4 OUTER ECCENTRIC-A
MAIN
1 INPUT SHAFT
2 INNER ECCENTRIC
8 OUTPUT CAGE-A
INPUT

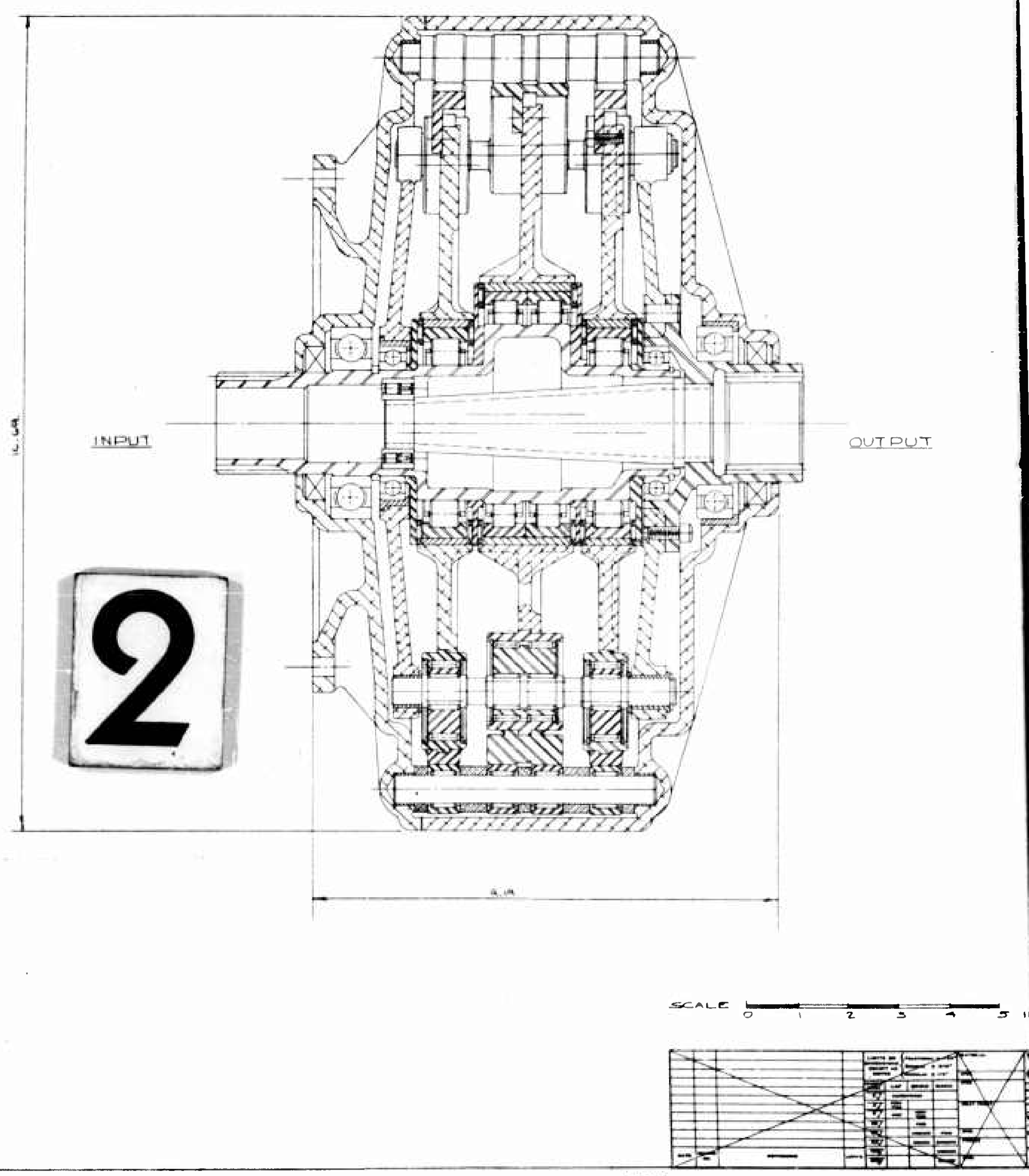
1

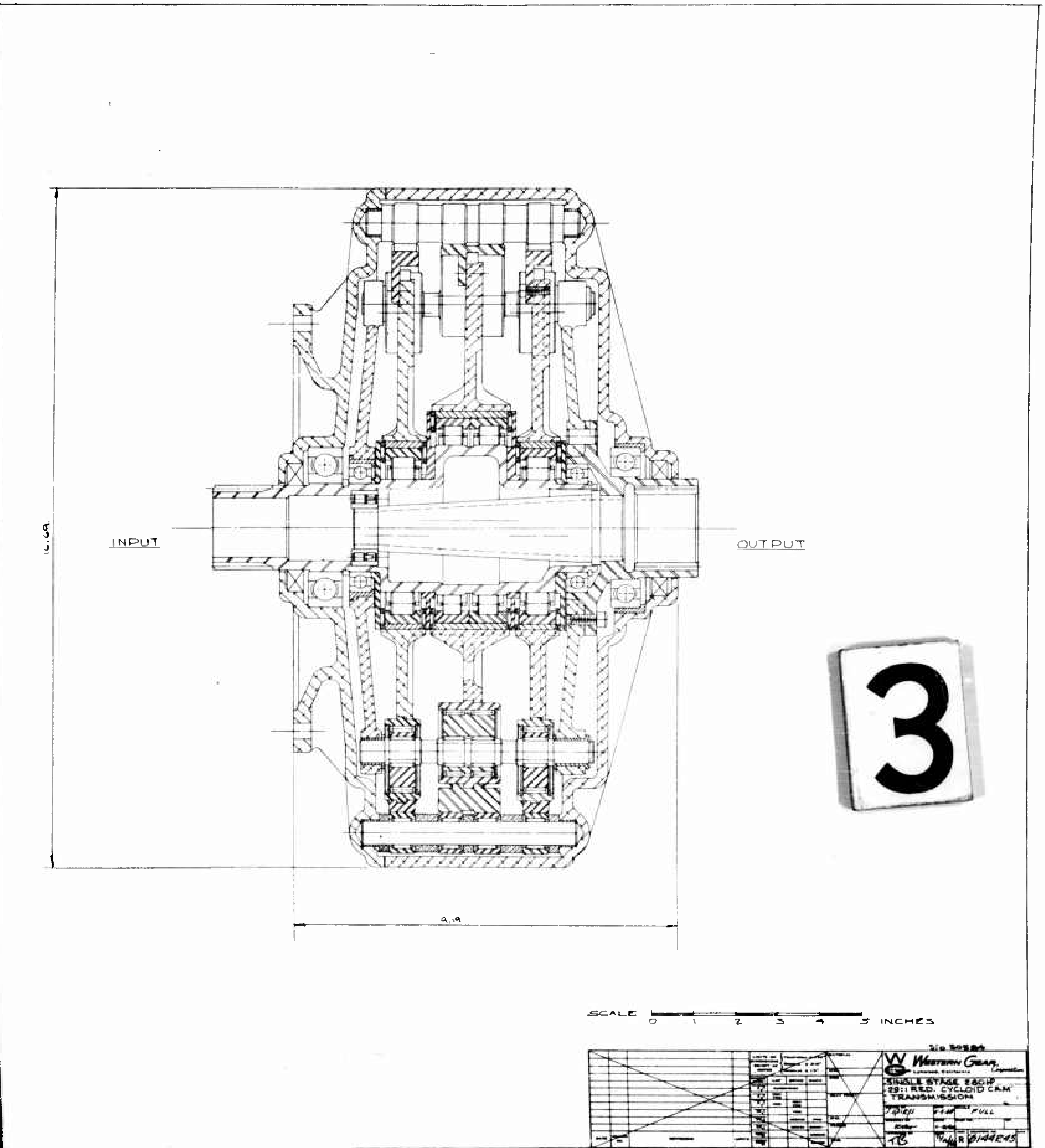


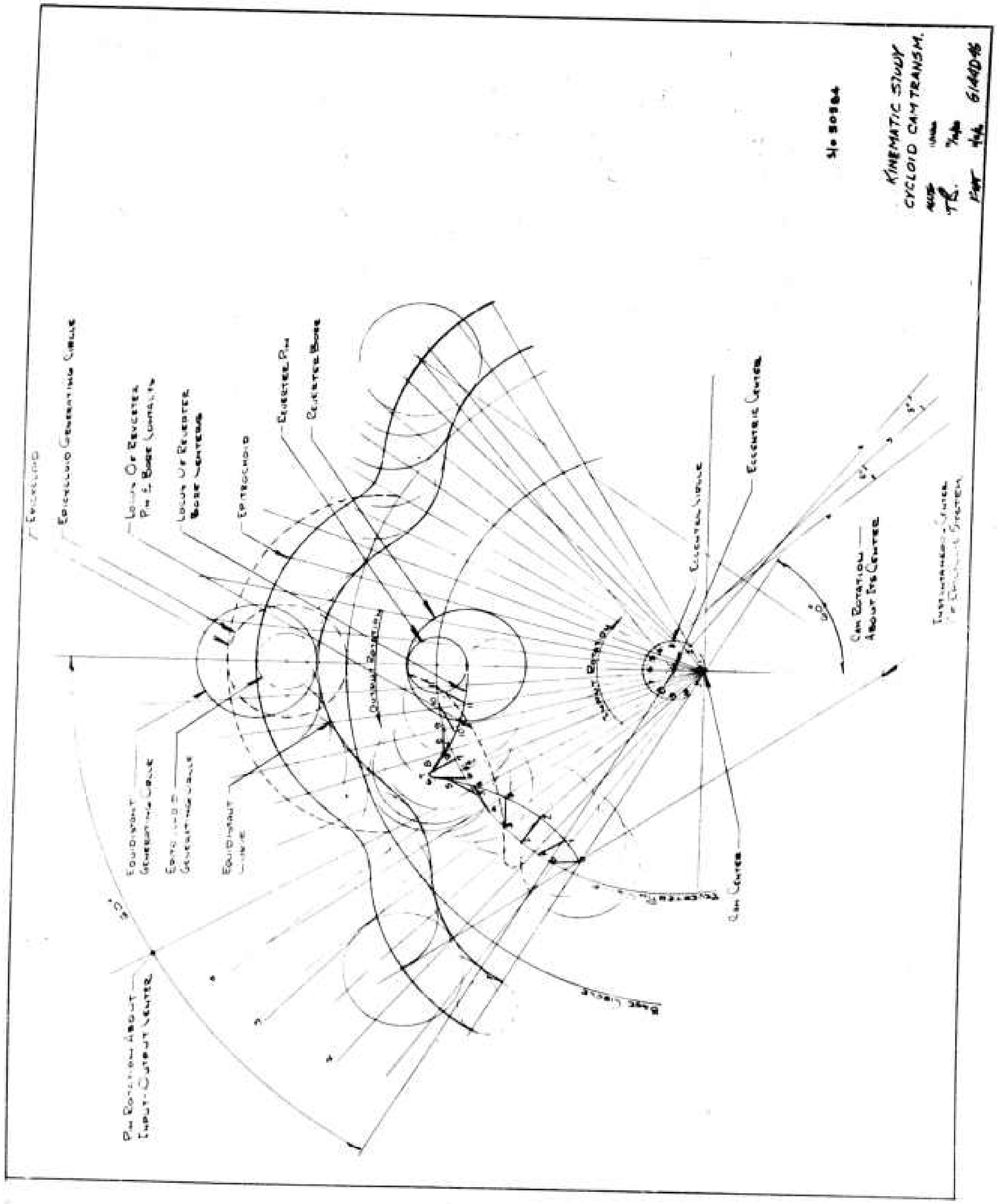
REVERTING ECCENTRIC-A
REVERTING ECCENTRIC-INNER
REVERTING ECCENTRIC-B
REVERTER PINS
POWER INPUT GEAR
4 OUTER ECCENTRIC-A
POWER OUTPUT GEAR
MAIN
AXIS
15.15
1 INPUT SHAFT
2 INNER ECCENTRIC
3 OUTER ECCENTRIC-B
6 OUTER CAM PLATE-B
5 INNER CAM PLATE
7 OUTER CAM PLATE-A
8 OUTPUT CAGE-A
9 OUTPUT CAGE-B
REACTION PINS/ROLLERS
CONTAINED IN FIXED HOUSING
INPUT
POWER FLOW
OF THIS LAYOUT
OUTPUT
SCALE
0 1 2 3 4 5 INCHES
2
WESTERN GEAR
30:1 REDUCTION - 19:1
CYCLOID STAGE
FULL
6144R43



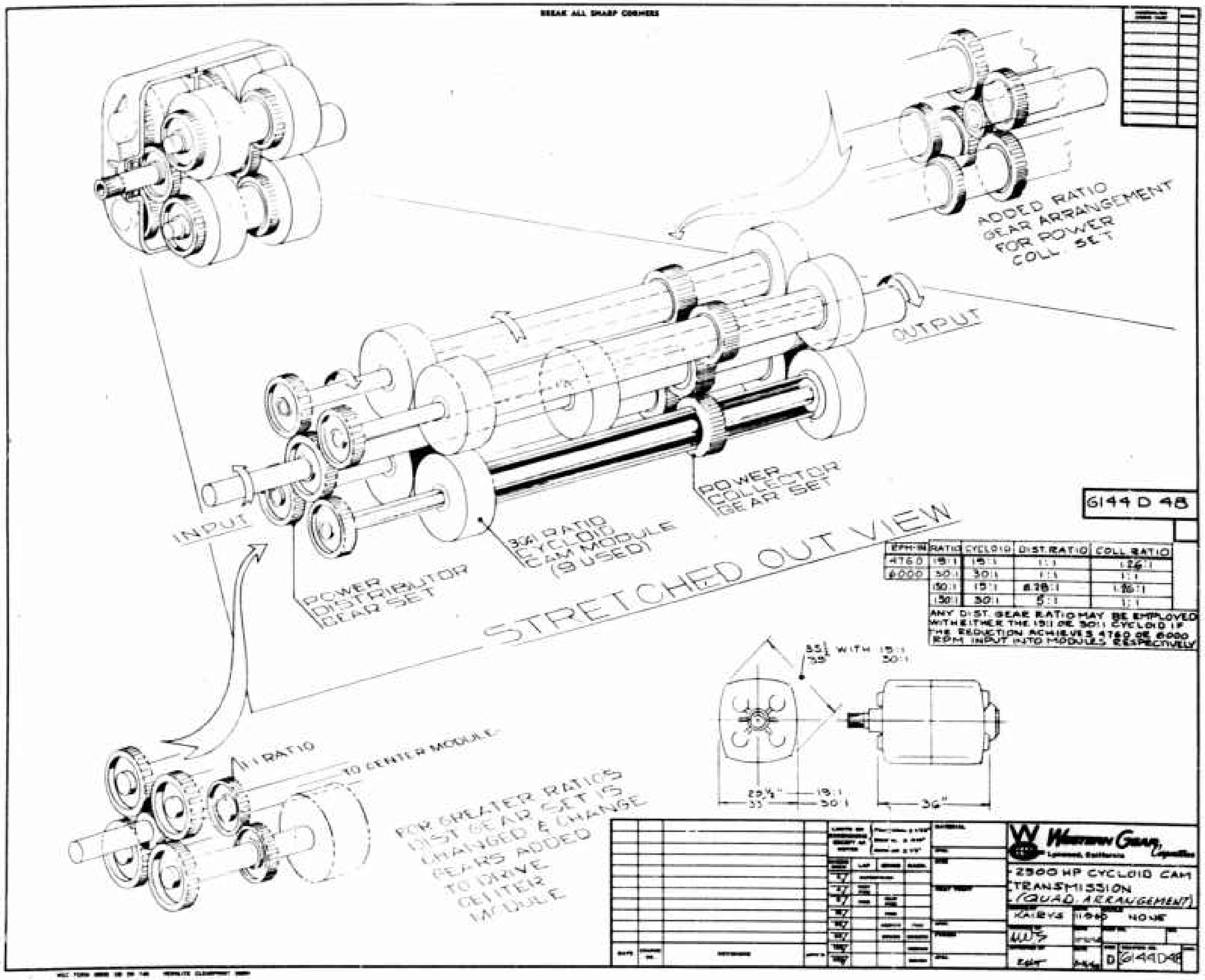
1

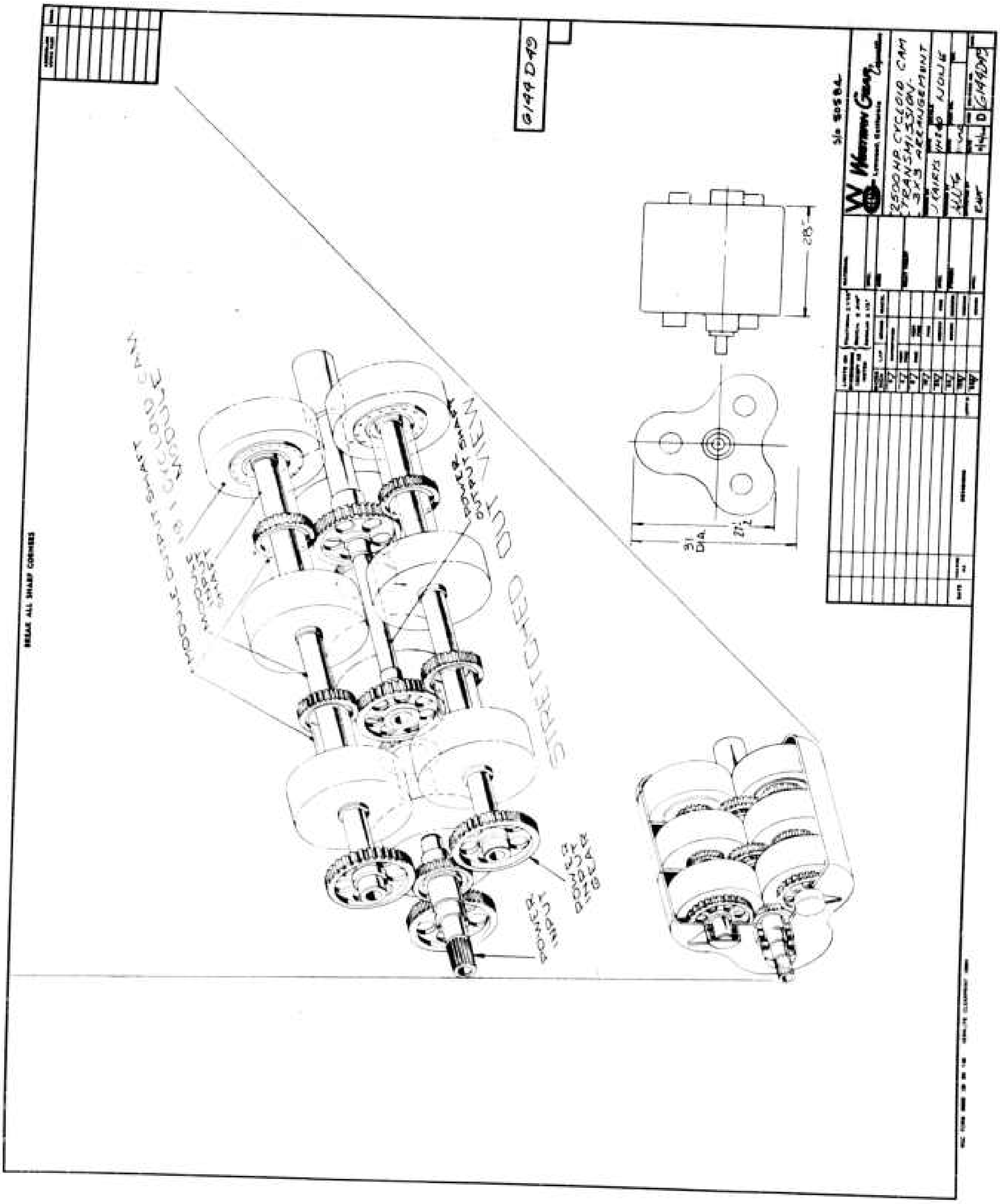

INPUT
OUTPUT
2
SCALE
0 1 2 3 4 5 IN


INPUT
OUTPUT
16.69
9.19
3
SCALE
0 1 2 3 4 5 INCHES
Western Gear Corporation
SINGLE STAGE 280HP
29:1 RED. CYCLOID CAM
TRANSMISSION
FULL
R
6144E45

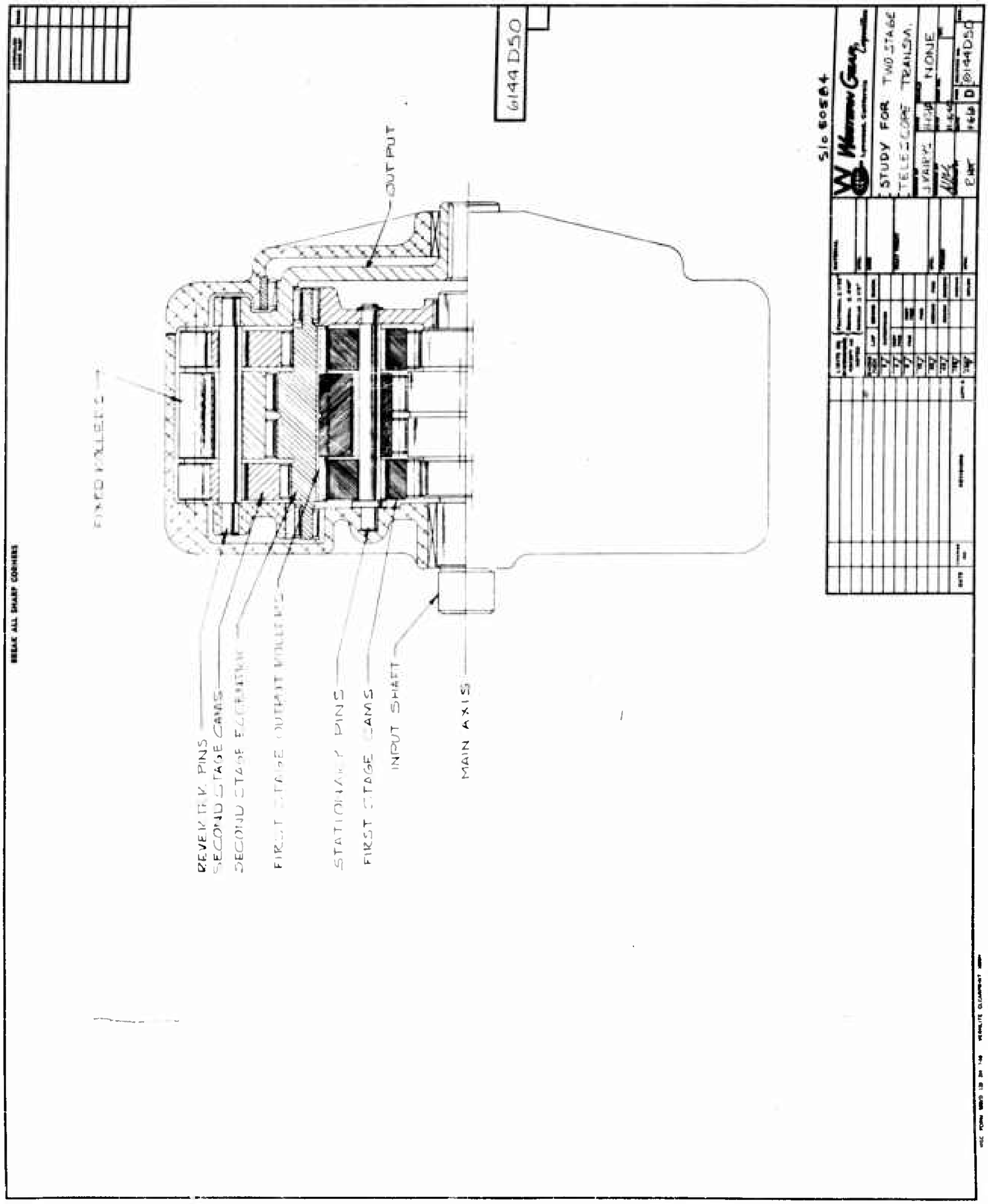


Epicycloid
Epicycloid Generating Circle
Locus Of Reducer Pin & Bore Contacts
Locus Of Reducer Bore Centers
Epitrochoid
Reducer Pin
Reducer Bore
Pin Rotation About Input-Output Center
Equidistant Generating Circle
Equidistant Curve
Output Rotation
Input Rotation
Eccentric Circle
Base Circle
Cam Center
Eccentric Center
60°
Cam Rotation About Its Center
Instantaneous Center Of Epicyclic System
S/o 50584
KINEMATIC STUDY
CYCLOID CAM TRANSM.
TR.
614AD16

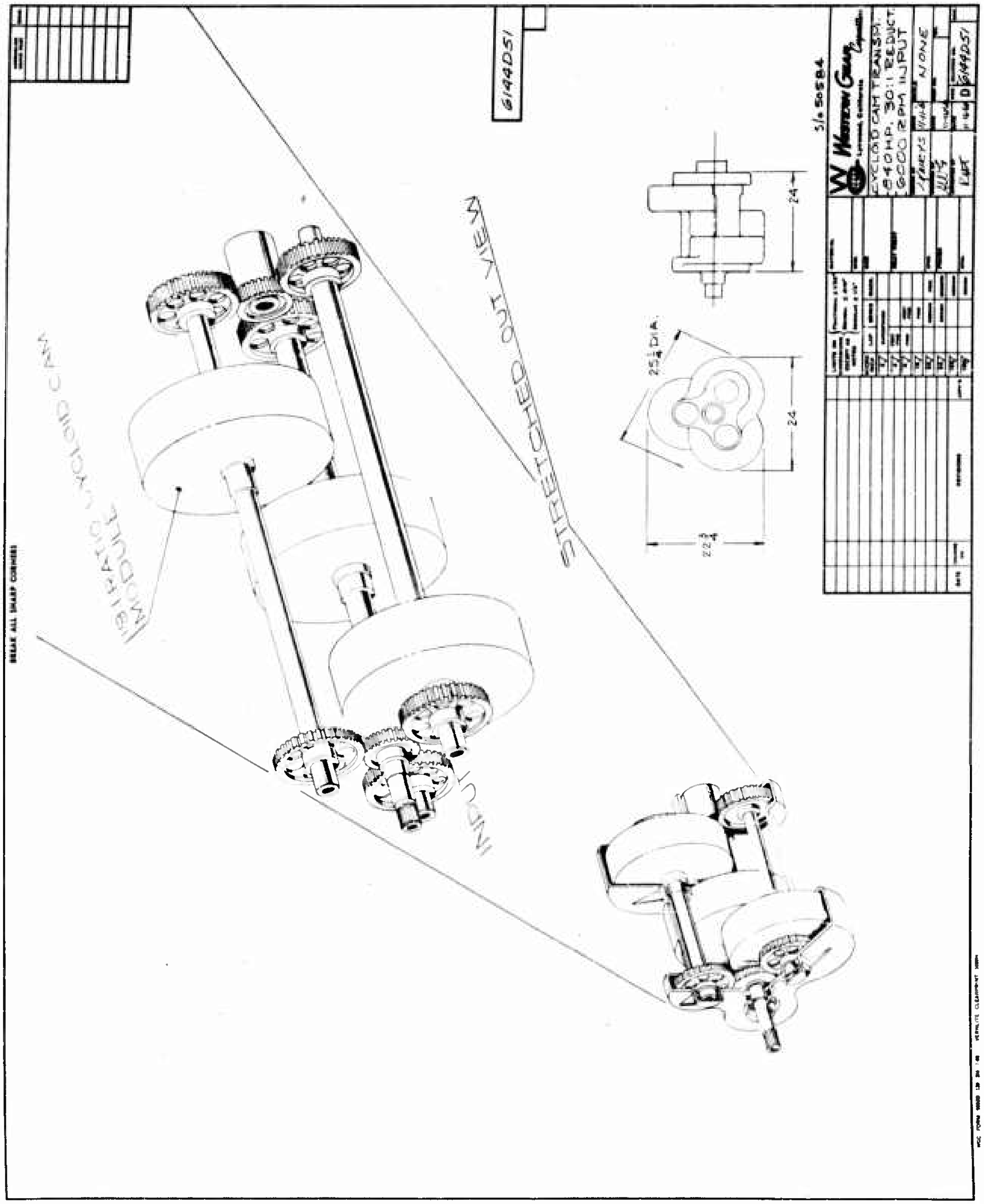

BREAK ALL SHARP CORNERS
ADDED RATIO GEAR ARRANGEMENT FOR POWER COLL SET
OUTPUT
INPUT
POWER COLLECTOR GEAR SET
30:1 RATIO CYCLOID CAM MODULE (9 USED)
POWER DISTRIBUTOR GEAR SET
STRETCHED OUT VIEW
6144 D 48
RPM IN | RATIO | CYCLOID | DIST. RATIO | COLL RATIO
4760 | 15:1 | 15:1 | 1:1 | 1.26:1
6000 | 30:1 | 30:1 | 1:1 | 1:1
ANY DIST. GEAR RATIO MAY BE EMPLOYED WITH EITHER THE 15:1 OR 30:1 CYCLOID IF THE REDUCTION ACHIEVES 4760 OR 6000 RPM INPUT INTO MODULES RESPECTIVELY
WITH 15:1
30:1
36"
1:1 RATIO
TO CENTER MODULE
FOR GREATER RATIOS DIST GEAR SET IS CHANGED & CHANGE GEARS ADDED TO DRIVE CENTER MODULE
Western Gear Corporation
2500 HP CYCLOID CAM TRANSMISSION (QUAD ARRANGEMENT)
NONE
D 6144D48

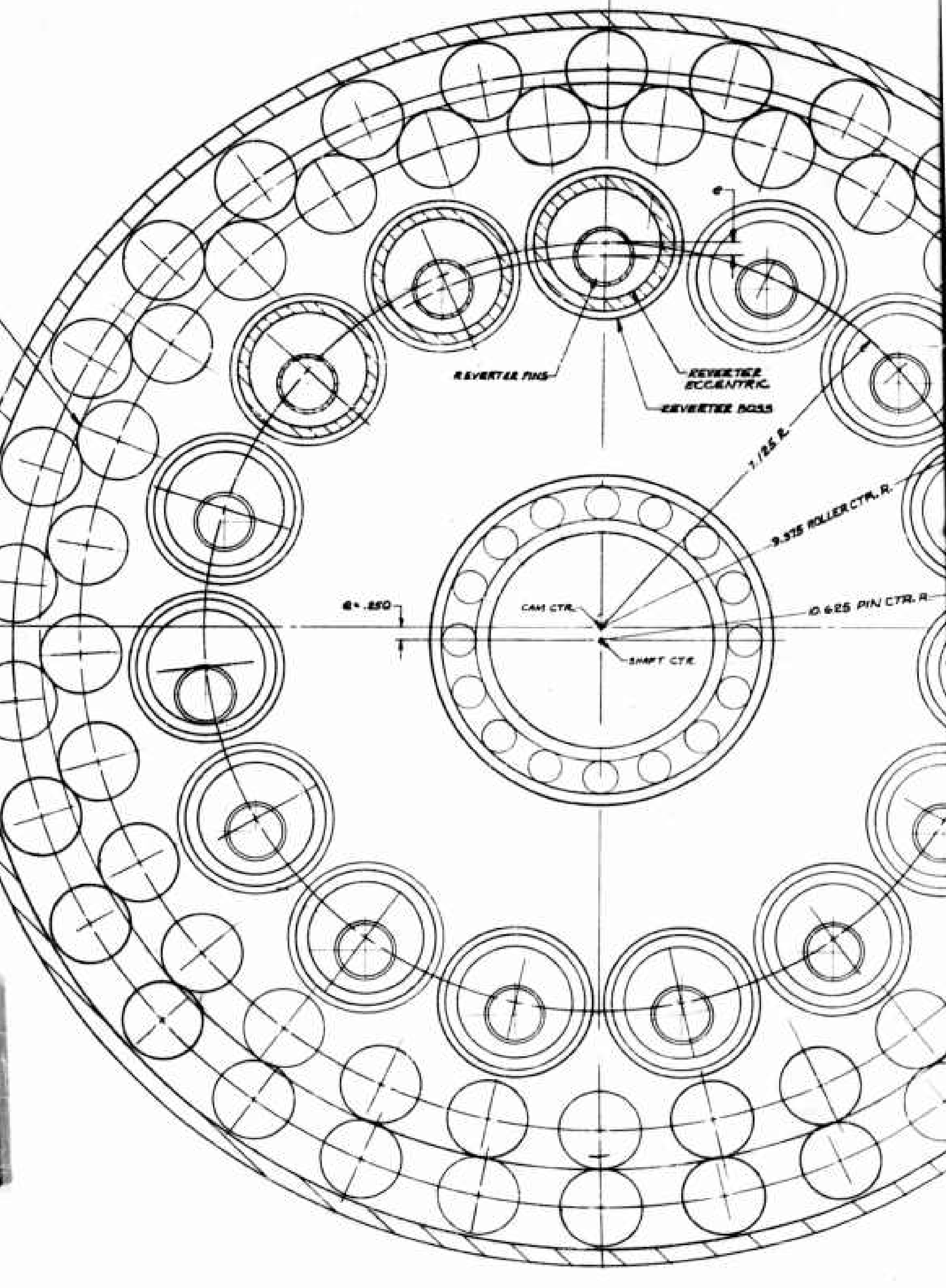


STRETCHED OUT VIEW
POWER INPUT
POWER OUTPUT GEAR
POWER & OUTPUT SHAFT
G144D49
2500 HP. CYCLOID CAM TRANSMISSION 3X3 ARRANGEMENT
31 DIA.
28"
BREAK ALL SHARP CORNERS

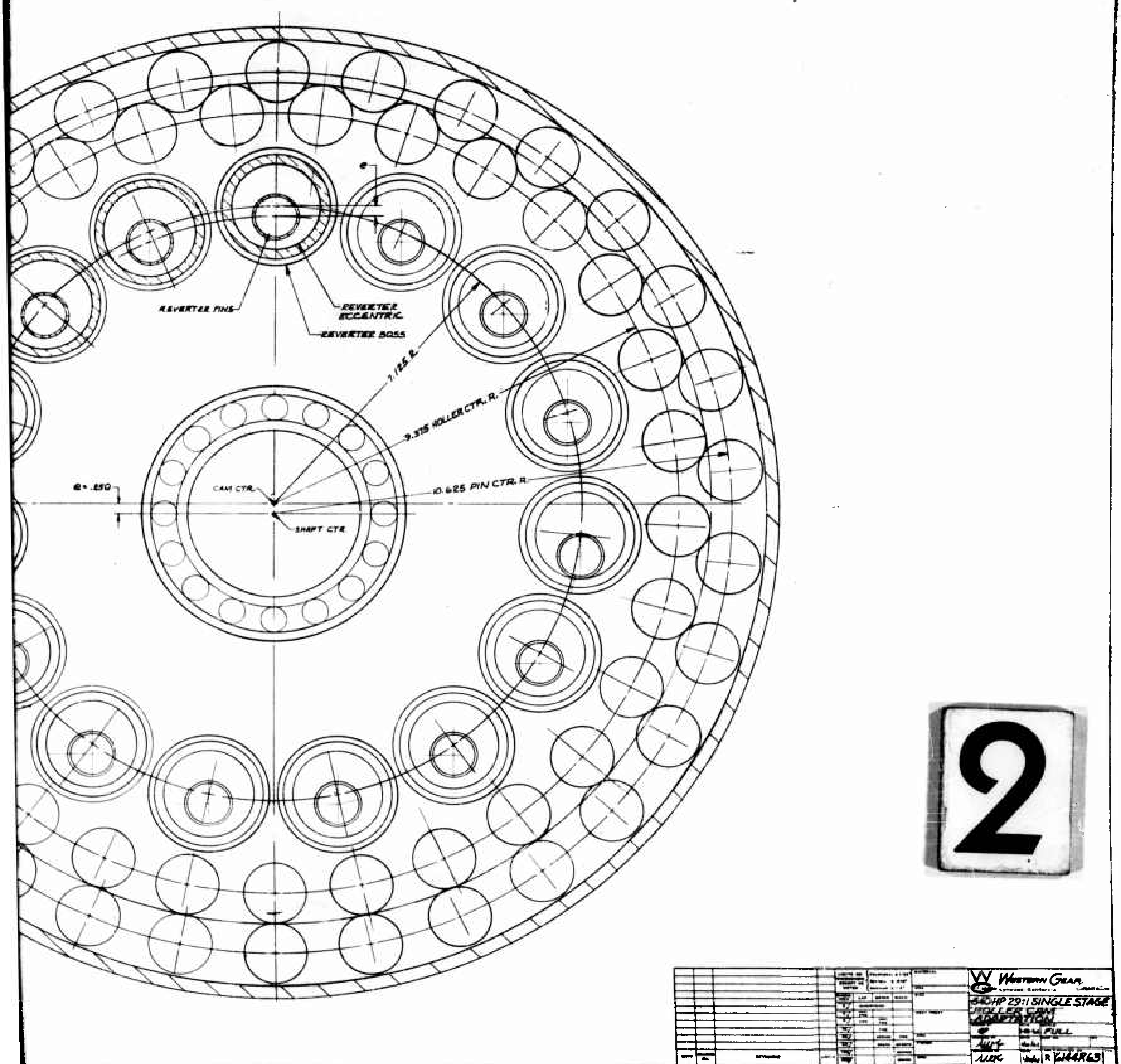


OUT PUT
FIXED ROLLERS
SECOND STAGE CAMS
FIRST STAGE OUTPUT ROLLERS
STATIONARY PINS
FIRST STAGE CAMS
INPUT SHAFT
MAIN AXIS
6144 D50
STUDY FOR TWO STAGE TELESCOPE TRANSM.
NONE
BREAK ALL SHARP CORNERS

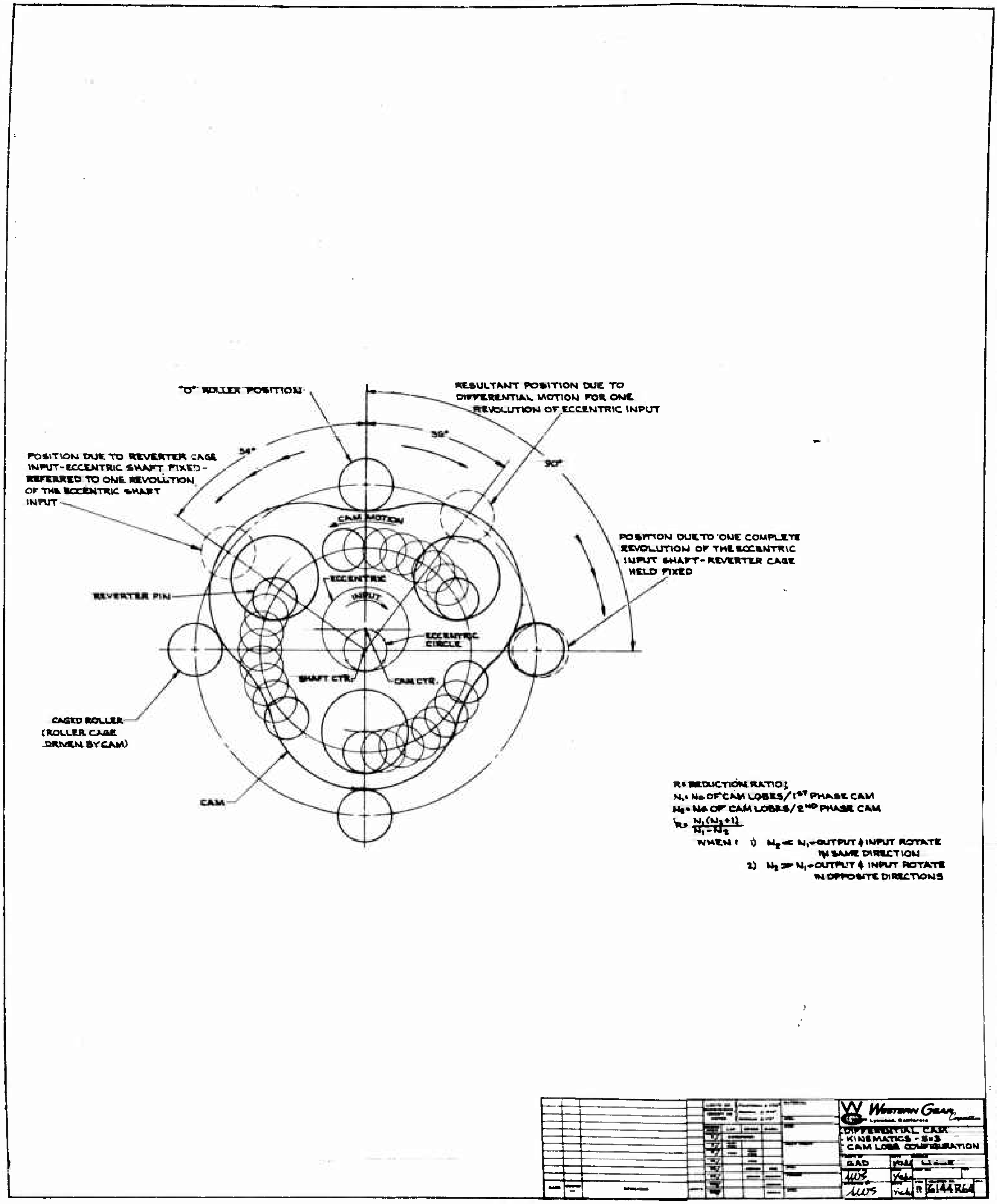

19:1 RATIO CYCLOID CAM MODULE
STRETCHED OUT VIEW
INPUT
BREAK ALL SHARP CORNERS
6144D51
S/o 50584
CYCLOID CAM TRANSM.
840 HP. 30:1 REDUCT.
6000 RPM INPUT
NONE
24
25 1/4 DIA
22 3/4


REACTION ROLLER
REACTION PIN
HOUSING
24.500 O.D.
REVERTER PINS
REVERTER ECCENTRIC
REVERTER BOSS
e
1.125 R.
9.375 ROLLER CTR. R.
10.625 PIN CTR. R.
e = .250
CAM CTR.
SHAFT CTR.

1



REVERTER PINS
REVERTER ECCENTRIC
REVERTER BOSS
7.125 R.
9.375 ROLLER CTR. R.
10.625 PIN CTR. R.
e = .250
CAM CTR.
SHAFT CTR.
2
WESTERN GEAR
HP 29:1 SINGLE STAGE
ROLLER CAM
ADAPTATION
FULL
R 6144R63



"O" ROLLER POSITION
RESULTANT POSITION DUE TO DIFFERENTIAL MOTION FOR ONE REVOLUTION OF ECCENTRIC INPUT
54°
36°
90°
POSITION DUE TO REVERTER CAGE INPUT-ECCENTRIC SHAFT FIXED - REFERRED TO ONE REVOLUTION OF THE ECCENTRIC SHAFT INPUT
CAM MOTION
POSITION DUE TO ONE COMPLETE REVOLUTION OF THE ECCENTRIC INPUT SHAFT - REVERTER CAGE HELD FIXED
ECCENTRIC
INPUT
REVERTER PIN
ECCENTRIC CIRCLE
SHAFT CTR.
CAM CTR.
CAGED ROLLER (ROLLER CAGE DRIVEN BY CAM)
CAM
R = REDUCTION RATIO;
N_1 = No. OF CAM LOBES/1ST PHASE CAM
N_2 = No. OF CAM LOBES/2ND PHASE CAM
$R = \frac{N_1(N_2+1)}{N_1-N_2}$
WHEN: 1) $N_2 < N_1$ - OUTPUT & INPUT ROTATE IN SAME DIRECTION
2) $N_2 > N_1$ - OUTPUT & INPUT ROTATE IN OPPOSITE DIRECTIONS
WESTERN GEAR Corporation
Lynwood, California
DIFFERENTIAL CAM KINEMATICS - 3x3 CAM LOBE CONFIGURATION
GAD
R
6144R64

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Liaison Group, Room 208
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1. Aircraft Components

Western Gear Corporation, 2600 East Imperial Highway, Lynwood, California, CYCLOIDAL CAM TRANSMISSION - Systems Management Division, Western Gear Corporation.

TCREC Tech Rept. 61-38, June 1961,271pp. (Task 9R38-01-020-04)

Unclassified Report

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